ROPER'S HAND-BOOK OF LAND AND MARINE ENGINES.

OPINIONS OF THE PRESS.

Iron Age, New York.

THE author of this hand-book says, in his preface, that his object in preparing it, "has been to present to the practical inquirer a book to which he can refer with confidence for information in regard to every branch of his profession."

Rules and directions expressed in algebraic formulæ are of little service to the majority of engineers, because they are not fully understood. The author, keeping this in mind, has avoided most of the points which render many of our hand-books of limited value to the practical man. He has had a long and extensive practical experience among the men for whom he writes, and understanding their wants, has produced a book which seems admirably adapted to those who have anything to do, in a practical way, with steam machinery. We have given the work a careful examination, and consider it one of the most satisfactory works of the kind we have ever seen. Mr. Roper thoroughly understands his subject, being entirely practical, and, at the same time, having a correct understanding of scientific principles. His chapters on the theory of steam engineering are so simple and practical that there is no mechanic in the country, however ignorant he may be of higher mathematics, who cannot learn all they are intended to teach. His practical directions for the management of engines are just such as we should expect from an experienced engineer who had spent all his life in an engine-room, but who had learned the theory as well as the practice of his trade. They are plain and to the point, and the reader may accept them with an entire confidence. His descriptions of engines, pumps, and the appliances connected with engines, are exceedingly satisfactory, as are also his rules, which seem to be the best and simplest which could be formulated. The book has an abundance of tabular information, which seems to include all the tables that could be of any use. The engravings are good, and are just what is wanted to explain the text. In a word, the amount and kind of information contained in this work seems to be all that could be desired. The owner of a steam-engine cannot well do without it, and no one who runs an engine should be ignorant of any part of its contents.

American Artisan, N. Y.

THIS book is an exhaustive treatise on land and marine engines, and the facts it contains are arranged with general headings and side headings along the pages, so as to be referred to with the utmost facility. An immense amount of useful Tables to assist in computation find place in the book; simple theories for various calculations are also a very important feature in its arrangement. The information it gives contains very little purely theoretical discussion, but consists of clear and brief statements of facts, methodically arranged. It will prove a valuable assistant to any engineer on land or water, and cannot fail to have an immense sale. It is bound in pocket-book form, to be carried in the pockets of engineers for convenient and ready reference.

Locomotive Engineers' Monthly Journal, Cleveland, Ohio.

WE are just in receipt of this excellent book, a work of 600 pages, and from a careful examination with a view to satisfy ourselves as to its worth, we freely say that Mr. Roper has comprehended exactly what is needed in a "hand-book." It abounds in plates illustrating and aiding to understand what is fully explained in regard to the working parts of all classes of steam-engines, their relation to each other, the proper method of "setting up," and in fact there is hardly a question connected with steam and its application to machinery that is not considered and ably treated. Another feature of the book is its carefully prepared rules and tables which are constantly needed by engineers, machinists, and all others whose duty it is to either build or manage machinery. As its title indicates, its form is such that it can be carried in the pocket and referred to at any time and under any circumstances, and this fact adds largely to its value.

The American Engineer, Baltimore, Md.

WE have received a copy of Roper's Hand-Book of Land and Marine Engines. Our knowledge of the large experience of the author entitled us to expect from him a valuable contribution to the engineering literature of the class referred to, and an inspection of the plan and contents of the "Hand-Book" shows its value at once. The most practical information upon all subjects interesting to the practical engineer, is given in the plainest and most common-sense manner. The absence of algebraical formulæ is an agreeable feature of the work to practical engineers. The matters expressed algebraically in ordinary books of this class, are here expressed in all the simplicity of the printer's art. The most modern views are found to pervade the book, and in this respect it differs from other hand-books which have retained popularity too long. We commend the book to all practical engineers.

Scientific American, New York.

THE house of Messrs. E. CLAXTON & COMPANY have just published a new work on "Land and Marine Engines," by Stephen Roper, Engineer, author of "Roper's Catechism of High-Pressure or Non-Condensing Steam-Engines," and "Roper's Hand-Book of the Locomotive," etc. Mr. Roper needs no introduction to our readers as a competent and trustworthy authority on steam engineering, and his present volume will prove exceedingly valuable to all engineers who desire a treatise combining scientific accuracy with a popular style — free from formulæ and ultra-mathematical expressions. The Tables with which the work is interspersed are numerous and valuable, and there is at the end of the book a very interesting historical account of the invention and improvement of the steam-engine.

Trade Journal, Philadelphia.

THIS is an elegant, large duodecimo volume, of about 600 pages, in flexible morocco binding, recently issued by Messrs. E. CLAXTON & CO., of this city. It is just what it professes to be, a hand-book, not an exhaustive treatise on any particular branch of engineering. It contains as large a number of good illustrations as could be expected in a work of its size; and most of these are evidently designed to furnish information rather than to gratify curiosity. It appears to be especially rich in practical information, while it does not ignore or cover up those philosophical principles on which the engineer's success depends. It not only tells how engines of the various kinds are built, but how to manage and take care of them, and how to read the signs of deterioration, and determine what ails them when they are out of order. Mr. Roper has given descriptions with more or less detail of quite a number of steam-engines, both high and low pressure, and of various governors, cut-offs, injectors, indicators, and other appendages to the steam-engine. This must have been a very delicate part of his work; especially as many of these were selected from among a great number of rival machines, and we must do him the justice to say that he has shown nothing like partisan feeling; nor, as far as we can perceive, even partiality for one or prejudice against another in the management of this difficult part of his work. We believe that this hand-book will be a great convenience to professional engineers, on account of its convenient size, low cost, and plainness of speech.

Manufacturer, Newark.

TO the constructive mechanical literature of the age, Mr. Stephen Roper, of Philadelphia, Pa., has recently added an important work, entitled "Hand-Book of Land and Marine Engines," as valuable a book of reference for those for whom it is published as was "Roper's Hand-Book of the Locomotive," which met with so much

favor at the hands of locomotive engineers. Like the latter, it is bound in pocket-book form, to be carried in the pockets of engineers for convenient reference. The book consists of clear and brief statements of facts, methodically arranged. It cannot fail to be a valuable assistant to any engineer on land or water, and manufacturers of steam-engines would probably find it to their advantage to furnish a copy of this book to every engineer where their engines are in use.

The Locomotive, Hartford, Conn.

A GREAT many books have been written since the days of Watt upon the theory and practical working of the steam-engine. Some have gone so far into the realms of theory that few, besides professional men, could understand them, while others, in endeavoring to adapt their writings to practical men only, have stripped the subject of scientific interest, and rendered it altogether too elementary. What has been wanted is a series of books upon the steam-engine which, while explaining its operations, should happily combine *theory and practice* in such a way as to commend itself to professional engineers, and at the same time be within the comprehension of *practical* men in the engine-room.

It is well understood that such a work could only be accomplished by a man who was practically familiar with all kinds of steam-engines, and, at the same time, well up in the theory. Stephen Roper, of Philadelphia, being impressed with the fact that such information was called for, undertook the task of providing it. He has brought out three books, for which engineers throughout the country are laid under the greatest obligation. The "Catechism of the Steam-Engine," "The Hand-Book of the Locomotive," and "Hand-Book of Land and Marine Engines," are the names of Mr. Roper's publications. The last, "Hand-Book of Land and Marine Engines," is exceedingly complete, and should be in the hands of every practical engineer. We commend these works, one and all, to the engineers of the country.

National Car Builder, New York.

HAND-BOOK of Land and Marine Engines," by Stephen Roper, Engineer, author of several valuable works on steam mechanism. The present volume contains 600 pages, and is very compactly and handsomely bound, and will prove extremely useful to operatives and others who desire a clear, intelligible, and practical treatise on the modelling, construction, running, and management of land and marine engines and boilers. The work contains numerous illustrations, a number of valuable tables, and a full and well-arranged index. The reputation of the author as a competent and trustworthy authority on steam engineering will doubtless insure for this publication an extensive and ready sale.

ROBERT FULTON.

A NAME WE SHOULD NEVER FORGET TO HONOR.

FRONTISPIECE.

HAND-BOOK

OF

Land and Marine Engines,

INCLUDING

THE MODELLING, CONSTRUCTION, RUNNING, AND MANAGEMENT OF LAND AND MARINE ENGINES AND BOILERS.

With Illustrations.

BY

STEPHEN ROPER, Engineer,

Author of

"Roper's Catechism of High-Pressure or Non-Condensing Steam-Engines," "Roper's Hand-Book of the Locomotive," "Roper's Hand-Book of Modern Steam Fire-Engines," "Roper's Handy-Book for Engineers," "Roper's Improvements in Steam-Engines," "Roper's Use and Abuse of the Steam-Boiler," "Roper's Questions and Answers for Engineers," etc.

Seventh Edition, Revised, Enlarged and Improved.

PHILADELPHIA:

E. CLAXTON & COMPANY,

930 Market Street.

1883.

J. FAGAN & SON,
ELECTROTYPERS, PHILAD'A.

TO THE

ENGINEERS OF THE UNITED STATES,

THIS BOOK

Is Respectfully Inscribed.

"Steam Engineering is one of the noblest sciences that ever attracted the attention of man."

"A place for everything and everything in its place."—See page 264.

INTRODUCTION.

THE object of the writer in preparing this work has been to present to the practical engineer a book to which he can refer with confidence for information regarding every branch of his profession. Many of the books heretofore written on this subject are full of formulæ for calculating questions that may arise in the engine-room; but, as they are generally expressed in algebraical form, they are of little service to the majority of engineers; for, however useful such formulæ may be to the scientific, they can be of no practical value to men who do not fully understand them. It is also no less a fact that nearly all writers on the steam-engine deal more with the past than the present. This is to be regretted, for, however interesting the bygone records of steam engineering may be, *as* a history, they cannot instruct the engineer of the present day in the principles and practice of his profession.

An experience of over thirty years with all kinds of engines and boilers enables the writer to fully comprehend the wants of the class for whom he writes, and what they can understand and employ. With this object in view, he has carefully investigated all the details of Land and Marine Engines and Boilers, taking up each subject singly, and excluding therefrom everything not directly connected with Steam Engineering. Particular attention has been given to the latest improvements in all classes of engines,

and to their proportions according to the best modern practice, which will be found of special value to engineers, as nothing of the kind has heretofore been published. The book also contains ample instructions for setting up, lining, reversing, and setting the valves of all classes of engines, — subjects that have not, up to the present time, received that attention from writers on the steam-engine that their importance to engineers so justly merits. It also contains a complete Lexicon of Central, Mechanical, and Natural Forces, which will be found of great value to engineers, as scarcely anything of the kind that has heretofore been published, has been applicable to American practice. A large portion of the work is devoted to the examination and discussion of the principles of Hydro- and Thermodynamics, which include Air, Water, Heat, Combustion, Steam, Liquefaction, Dilatation of Gases, Molecular and Atomic Forces, Dynamic Equivalents, subjects with which the practical engineer should be fully conversant; as to ignore the principles of any subject is similar to building a structure without knowing the strength of the foundation; for it was only by a minute and careful analysis of the physical phenomena which convert heat into a motor force that the steam-engine has been brought to its present perfection. The strength of materials, design, construction, care, and management of all classes of Steam-Boilers are also fully discussed. In the preparation of the work, the writer consulted Auchincloss, Weissenborn, Buel, Nystrom, and the "Scientific American," for which he makes due acknowledgment.

The writer candidly admits that the work may be found somewhat defective in language, but he firmly believes that it will be found perfectly accurate and reliable in all other respects.

S. R.

CONTENTS.

For a full reference to the Contents in detail, see Index, page 583.

	PAGE
INTRODUCTION	7
THE STEAM-ENGINE	21
STEAM	25
Table showing the Temperature and Weight of Steam at different Pressures from 1 Pound per Square Inch to 300 Pounds, and the Quantity of Steam produced from 1 Cubic Inch of Water, according to Pressure	39
WORKING STEAM EXPANSIVELY	43
Table of Hyperbolic Logarithms to be used in Connection with the above Rule	48
Table showing the average Pressure of Steam upon the Piston throughout the Stroke, when Cut-off in the Cylinder from $\frac{1}{3}$ to $\frac{7}{11}$, commencing with 25 Pounds and advancing in 5 Pounds up to 75 Pounds Pressure	49
Table showing the average Pressure of Steam upon the Piston throughout the Stroke, when Cut-off in the Cylinder from $\frac{1}{3}$ to $\frac{7}{9}$, commencing with 80 Pounds and advancing in 5 Pounds up to 130 Pounds Pressure	50
Table of Multipliers by which to find the mean Pressure of Steam at various points of Cut-off	51
HIGH-PRESSURE OR NON-CONDENSING STEAM-ENGINES	54
POWER OF THE STEAM-ENGINE	55

PAGE
Foreign Terms and Units for Horse-power . . . 59
Table of Factors 68
WASTE IN THE STEAM-ENGINE 70
DESIGN OF STEAM-ENGINES 73
THE BED-PLATE 74
CYLINDERS 74
Table showing the proper Thickness for Steam-cylinders of different Diameters 75
PISTONS 76
PISTON-RINGS 77
PISTON-SPRINGS 77
STEAM-PISTONS 78
SOLID PISTONS 78
Table of Piston Speeds for all Classes of Engines — Stationary, Locomotive, and Marine 79
PISTON, CONNECTING-ROD, AND CRANK CONNECTION . 80
Table showing the Position of the Piston in the Cylinder at different Crank-angles, according to the length of Connecting-rod 81
Table showing Length of Stroke and Number of Revolutions for different Piston Speeds in Feet per Minute . 82
PISTON-RODS 83
CRANK-PINS 83
Table showing the Angular Position of the Crank-pin corresponding with the various Points in the Stroke which the Piston may occupy in the Cylinder . . 84
STEAM-CHESTS 85
VALVE-RODS 85
GUIDES 85
ROCK-SHAFTS 86
CROSS-HEADS 86
STEAM-PORTS 87
Table showing the Proper Area of Steam-ports for different Piston Speeds 88
SLIDE-VALVES 89
PROPORTIONS OF SLIDE-VALVES 93

PAGE
Lap on the Slide-valve 93
Poppet or Conical Valves 95
Table showing the Amount of "Lap" required for Slide-valves of Stationary Engines when the Steam is to be Worked expansively 97
Lead of the Slide-valve 97
Clearance 99
Compression 100
Friction of Slide-valves 100
Balanced Slide-valves 102
Fitting Slide-valves 103
Slide-valve Connections 103
Eccentrics 104
Eccentric-rods 108
Cranks 109
Crank-shafts 114
Pillow-blocks, or Main Bearings 114
Fly-wheels 115
Link-motion 116
Proportions of Steam-engines according to the best Modern Practice 121
Setting up Engines 127
Dead-centre 128
How to put an Engine in Line 128
How to Reverse an Engine 131
Setting Valves 131
How to set a Slide-valve 132
Setting out Piston Packing 134
Piston- and Valve-rod Packing 135
Automatic Cut-offs 138
Governors 139
The Huntoon Governor 140
The Allen Governor 142
The Cataract 145
Wright's High-pressure Engine 146
Hawkins and Dodge's High-pressure Engine . 147

	PAGE
Watts and Campbell's High-pressure Engine	148
The Buckeye High-pressure Engine	148
Wheelock's High-pressure Engine	151
The Corliss High-pressure Engine	151
Hampson and Whitehill's High-pressure Engine	153
The Allen High-pressure Engine	155
Woodruff & Beach's High-pressure Engine	156
Naylor's Vertical High-pressure Engine	157
Williams' Vertical Three-cylinder High-pressure Engine	158
Roper's Caloric Engine	159
Haskins' Vertical High-pressure Engine	163
Massey's Rotary Engine	164
Portable Engines	166
How to balance Vertical Engines	166
Knocking in Engines	168
The Injector	168
Rue's "Little Giant" Injector	170
How to put on Letter "B" Injector	170
Method of working Letter "B" Injector	171
Table of Capacities of "Little Giant" Injector	172
Pumps	172
Force-pumps	174
Piston Pumps	175
Boiler Feed Pumps	176
Steam-pumps	178
The Atlas Steam-pump	178–179
The Dayton Cam-pump	180–181
Directions for setting up Steam-pumps	185
The Pulsometer	185
James Watt	187
Condensing, or Low-pressure Steam-engines	188
Explanation of the working Principles of the Condensing Engine	190
Horse-power of Condensing Engines	191
The Vacuum	193

	PAGE
MARINE STEAM-ENGINES	197
COMPOUND ENGINES	200
DIRECT-ACTING ENGINES	203
BALANCING THE MOMENTUM OF DIRECT-ACTING ENGINES	205
OSCILLATING ENGINES	205
TRUNK ENGINES	207
GEARED ENGINES	207
BACK-ACTION ENGINES	208
SIDE-LEVER ENGINES	209
BEAM ENGINES	209
MARINE BEAM ENGINE	211
STARTING-GEAR FOR MARINE ENGINES	215
CONDENSERS	216
AIR-PUMPS	222
THE HYDROMETER, SALINOMETER, OR SALT-GAUGE	223
THE MANOMETER	224
THE BAROMETER	225
MARINE ENGINE REGISTER, CLOCK, AND VACUUM GAUGE	226
STEAM-GAUGES	227
GLASS WATER-GAUGES	233
THE STEAM-ENGINE INDICATOR	235
TECHNICAL TERMS USED WITH THE INDICATOR	238
WHAT INDICATOR DIAGRAMS SHOW	242
INDICATOR DIAGRAMS	245
HOW TO KEEP THE INDICATOR IN ORDER	253
THE DYNAMOMETER	254
THE ENGINEER	256
MANAGEMENT OF LAND AND MARINE ENGINES	258
HOW TO PUT THE ENGINES IN A STEAMBOAT OR SHIP	265
SCREW-PROPELLERS	271
NEGATIVE SLIP OF THE SCREW-PROPELLER	274
Table of the Proper Proportions of Screw-propellers	279
MEASUREMENT OF THE SCREW-PROPELLER	279
HOW TO LINE UP A PROPELLER-SHAFT	282
PADDLE-WHEELS	282

PAGE
FLUID RESISTANCE 287
Signification of Signs used in Calculations . . . 291
DECIMAL 291
Decimal Equivalents of Inches, Feet, and Yards . . 292
Decimal Equivalents of Pounds and Ounces . . . 292
Useful Numbers in calculating Weights and Measures, etc. 292
Decimal Equivalents to the Fractional Parts of a Gallon or an Inch 294
UNITS 294
THEORY OF THE STEAM-ENGINE 298
WATER 299
Table showing the Weight of Water 305
Table showing the Weight of Water at different Temperatures 305
Table showing the Boiling-point for Fresh Water at different Altitudes above Sea-level 306
Table showing the Weight of Water in Pipe of various Diameters 1 Foot in Length 307
AIR 309
Table showing the Weight of the Atmosphere in Pounds, Avoirdupois, on 1 Square Inch, corresponding with different Heights of the Barometer, from 28 Inches to 31 Inches, varying by Tenths of an Inch 311
Table showing the Expansion of Air by Heat, and the Increase in Bulk in Proportion to Increase of Temperature 312
THE THERMOMETER 313
Comparative Scale of Centigrade, Fahrenheit, and Reaumer Thermometers 314
ELASTIC FLUIDS 320
CALORIC 321
HEAT 323
Latent Heat of Various Substances 331
Table of the Radiating Power of different Bodies . . 331

PAGE
Table showing the Effects of Heat upon different Bodies 332
COMBUSTION 332
Composition of different Kinds of Anthracite Coal . 336
Table showing the Total Heat of Combustion of various Fuels 342
Table showing the Nature and Value of several Varieties of American Coal and Coke, as deduced from Experiments by Professor Johnson, for the United States Government 343
Table showing some of the Prominent Qualities in the principal American Woods 344
Table showing the Relative Properties of good Coke, Coal, and Wood 344
Table of Temperatures required for the Ignition of different Combustible Substances 345
GASES 346
STEAM-BOILERS 350
STEAM-DOMES 353
MUD-DRUMS 355
SETTING BOILERS 356
EXPANSION AND CONTRACTION OF BOILERS . . . 358
TESTING BOILERS 359
NEGLECT OF STEAM-BOILERS 362
CARE AND MANAGEMENT OF STEAM-BOILERS . . 363
HEATING SURFACE 369
RULES FOR FINDING THE HEATING SURFACE OF STEAM-BOILERS 371
EVAPORATIVE EFFICIENCY OF BOILERS 372
HORSE-POWER OF BOILERS 375
FIRING 378
INSTRUCTIONS FOR FIRING 382
RULES FOR FINDING THE QUANTITY OF WATER BOILERS AND OTHER CYLINDRICAL VESSELS ARE CAPABLE OF CONTAINING 386
LONGITUDINAL AND CURVILINEAR STRAINS . . . 387

PAGE

RULES 388

EXPLANATION OF TABLES OF BOILER PRESSURES ON FOLLOWING PAGES 389

Table of safe Internal Pressures for Iron Boilers . . 390

Table of safe Internal Pressures for Steel Boilers . . 394

MARINE BOILERS 399

Proportions of Heating Surface to Cylinder and Grate Surface of noted Ocean, River, and Ferry-boat Steamers 402

SETTING MARINE BOILERS 404

BEDDING MARINE BOILERS 405

CLOTHING MARINE BOILERS 405

CARE OF MARINE BOILERS 406

REPAIRING STEAM-BOILERS 409

TUBES 410

Table of Superficial Areas of External Surfaces of Tubes of Various Lengths and Diameters in Square Feet 413

BOILER FLUES 417

Table of Squares of Thicknesses of Iron, and Constant Numbers to be used in finding the safe External Pressure for Boiler Flues 419

Table of safe Working External Pressures on Flues 10 Feet long 420

Table of safe Working External Pressures on Flues 20 Feet long 422

Table of Collapsing Pressure of Wrought-iron Boiler-flues $\frac{1}{4}$ Inch thick 425

Table of Collapsing Pressure of Wrought-iron Boiler-flues $\frac{5}{16}$ Inch thick 426

Table of Collapsing Pressure of Wrought-iron Boiler-flues $\frac{3}{8}$ Inch thick 427

Table of Collapsing Pressure of Wrought-iron Boiler-flues $\frac{7}{16}$ Inch thick 428

BOILER-HEADS 429

SAFETY-VALVES 431

PAGE

Table showing the Rise of Safety-valves, in Parts of an Inch, at different Pressures 433
Table of Comparison between Experimental Results and Theoretical Formulæ 435
RULES 437
FOAMING 439
INCRUSTATION IN STEAM-BOILERS 441
INTERNAL AND EXTERNAL CORROSION OF STEAM-BOILERS 449
BOILER EXPLOSIONS 452
COMPARATIVE STRENGTH, OF SINGLE- AND DOUBLE-RIVETED SEAMS 461
CALKING 464
STRENGTH OF THE STAYED AND FLAT SURFACES . . 468
DEFINITIONS AS APPLIED TO BOILERS AND BOILER MATERIALS 469
Table deducted from Experiments on Iron Plates for Steam-boilers, by the Franklin Institute, Phila. . 470
Table showing the result of Experiments made on different Brands of Boiler Iron at the Stevens Institute of Technology, Hoboken, N. J. 471
FEED-WATER HEATERS 472
Table showing the Units of Heat required to Convert One Pound of Water, at the Temperature of 32° Fah., into Steam at different Pressures 473
STEAM-JACKETS 474
LOSS OF PRESSURE IN CYLINDERS INDUCED BY LONG STEAM-PIPES 475
PRIMING IN STEAM-CYLINDERS 476
OILS AND OILING 477
Table of Coefficients of Frictions between Plane Surfaces 480
GRATE-BARS 482
CHIMNEYS 483
Table showing the proper Diameter and Height of Chimney for any kind of Fuel. 484

PAGE
SMOKE 485
MENSURATION OF THE CIRCLE, CYLINDER, SPHERE, ETC. 487
CENTRAL AND MECHANICAL FORCES AND DEFINITIONS 490
THE CIRCLE 511
Table containing the Diameters, Circumferences, and Areas of Circles, and the Contents of each in Gallons, at 1 Foot in Depth. Utility of the Table 511
LOGARITHMS 515
Table of Logarithms of Numbers from 0 to 1000 . . 516
HYPERBOLIC LOGARITHMS 517
Table of Hyperbolic Logarithms 518
Table containing the Diameters, Circumferences, and Areas of Circles from $\frac{1}{16}$ of an Inch to 100 Inches, advancing by $\frac{1}{16}$ of an Inch up to 10 Inches, and by $\frac{1}{8}$ of an Inch from 10 Inches to 100 Inches. . . 521
Table of Squares, Cubes, and Square and Cube Roots of all Numbers from 1 to 620 532
Table showing the Tensile Strength of various Qualities of Wrought-iron 547
Table showing the Actual Extension of Wrought-iron at various Temperatures 548
Table showing the Tensile Strength of various Qualities of Steel Plates 549
Table showing the Tensile Strength of various Qualities of Cast-iron 549
Table showing the Weight of Boiler-plates 1 Foot Square, and from $\frac{1}{16}$ to an Inch thick 551
Table showing the Weight of Square Bar-iron, from $\frac{1}{2}$ an Inch to 6 Inches Square, 1 Foot Long . . . 551
Table showing the Weight of Round-iron from $\frac{1}{2}$ an Inch to 6 Inches Diameter, 1 Foot Long . . . 552
Table showing the Weight of Cast-iron Balls from 3 to 13 Inches in Diameter 553

PAGE

Table showing the Weight of Cast-iron Plates per Superficial Feet as per Thickness 553

Table showing the Weight of Cast-iron Pipes, 1 Foot in Length, from $\frac{1}{4}$ Inch to $1\frac{1}{4}$ Inches thick, and from 3 to 24 Inches Diameter 554

Table showing the Weight per Square Foot of Wrought-iron, Steel, Copper, and Brass 555

RULES FOR FINDING THE DIAMETER AND SPEED OF PULLEYS 558

GEARING 559

BELTING 561

CEMENT FOR MAKING STEAM-JOINTS AND PATCHING STEAM-BOILERS 568

NON-CONDUCTORS FOR STEAM-PIPES AND STEAM-CYLINDERS 569

HOW TO MARK ENGINEERS' OR MACHINISTS' TOOLS . 569

TO POLISH BRASS 570

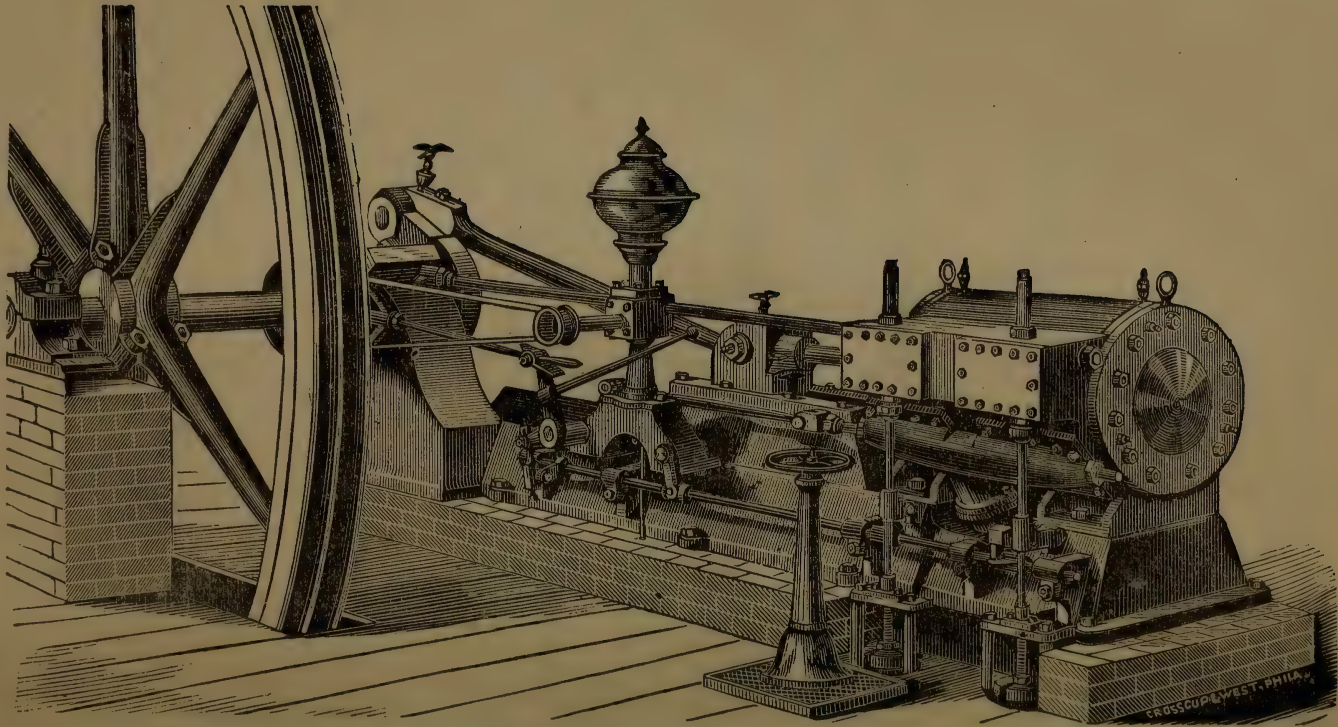
SOLDER 570

Table showing Weight of different Materials. . . 571

JOINTS 571

STEAM-BOILER FLUE AND TUBE CLEANERS . . . 573

THE INVENTION AND IMPROVEMENT OF THE STEAM-ENGINE 574



WRIGHT'S HIGH-PRESSURE CUT-OFF ENGINE.

HAND-BOOK

OF

LAND AND MARINE ENGINES.

THE STEAM-ENGINE.

NOTHING furnishes man with greater cause for congratulation, and even an excusable pride, than the feats of that mighty impersonation of brute force and human intellect,—the steam-engine, the Hercules of the nineteenth century,—which, once launched into the world's arena, has gone forth "conquering and to conquer," fulfilling its high destiny as a great civilizing agent, with an energy which no human arm can arrest and a rapidity which fills us with astonishment and admiration.

It would be superfluous here to attempt to enumerate the benefits which the steam-engine has conferred upon mankind. It is a matter of universal knowledge that all branches of industry have, since its introduction into use, made most important advances through its aid, and every day's experience shows it constantly extending its beneficial influence to new and important purposes, and lending its powerful assistance to the further advance of civilization.

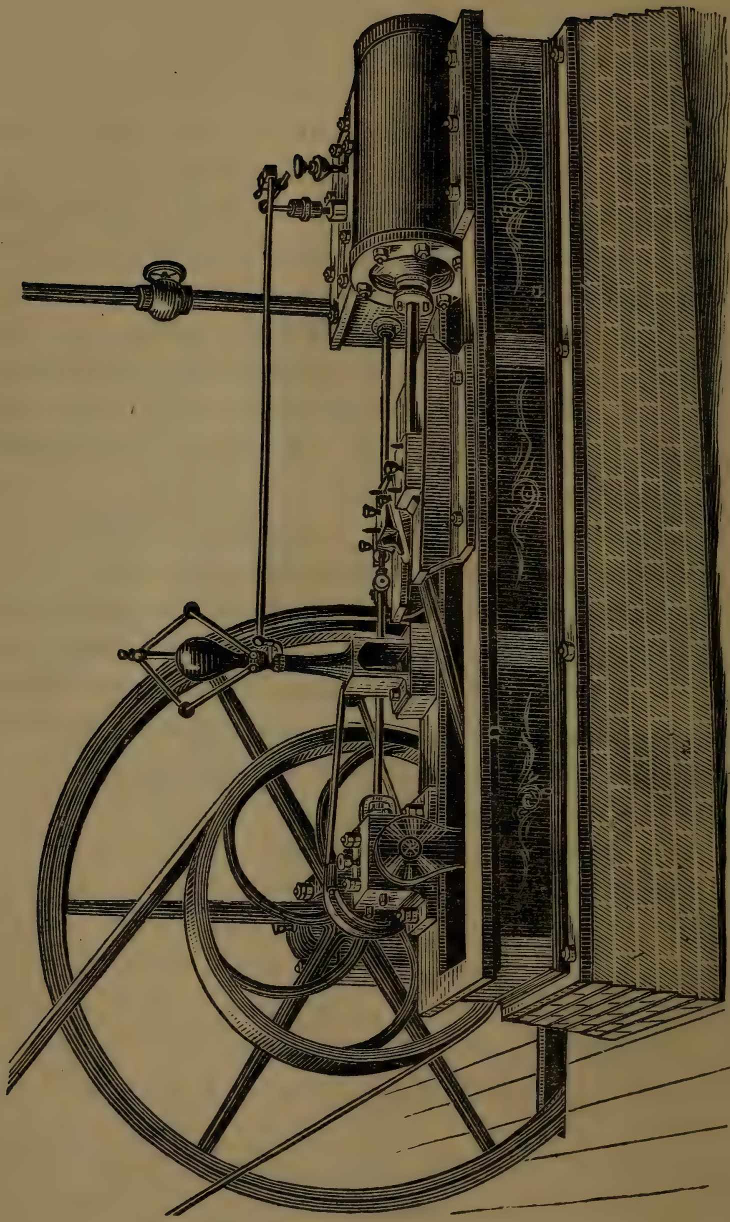
When we consider what the introduction of the steam-engine has already done, we have the less difficulty in anticipating that this invention may yet be destined to achieve objects of whose magnitude and importance we can at present form but a faint idea. There are few manufacturing processes that have not been revolutionized, simplified, and extended during the last fifty years through the agency of the steam-engine; but it is not alone in the large manufactory, the splendid steamer, and the rushing locomotive, that steam shows its usefulness, but also in our villages, cities, and towns, delving into the mines, driving the printing-press, helping all trades, and multiplying man's power a thousand-fold. Cities have sprung up under its magic touch, and on every side we see traces of the wonderful effects produced by this king of motors and mighty agent of civilization.

Then the independence of time, season, circumstance, and locality, in consequence of its not being influenced by flood, frost, winds, or drought, which mark the great superiority of this potent creation of engineering skill, and which, in its multiform applications and applicability, have invested it with an importance and an interest which success seems only to stimulate and render more intense, while its complexity of parts and diversity of combination offer a wide scope for the exercise of ingenuity, alike highly inviting to the theoretical and the practical mechanic.

Although the civilization of almost every people has been more or less affected by the introduction of steam, the extent to which steam-power is used is comparatively unknown. The total number of steam-engines of all descriptions in the world, at the close of eighteen hundred and seventy-four, was estimated at two hundred and seven thousand six hundred and seventy-seven, representing power equal to sixteen million horse-power, which would

be equivalent to the actual power of at least twenty-five millions of ordinary horses working night and day, or equal to the work of two hundred millions of men. Of this enormous steam-power the United States had three million nine hundred thousand horse-power, and Great Britain three million five hundred thousand. But it must not be supposed that the utilization of steam-power in the various productive industries of the world is circumscribing the area of manual labor or rendering it less remunerative than formerly; it is quite the reverse, as man profits, in some way, by every discovery in science; and though the discovery may compel him to abandon his former methods, it will still be found that the material results are invariably in his favor.

The steam-engine, even as a stationary power, is of recent origin; and contemplating the phases which it has already assumed, in connection with the general feeling that its energies have not yet been fully developed, it is not a matter of wonder that no other object in the entire range of human devices has so irresistibly arrogated to itself the devotion of the scientist and mechanic.



HAWKINS AND DODGE'S HIGH-PRESSURE THROTTLING ENGINE.

STEAM.

Steam is the elastic fluid into which water is converted by the continued application of heat.

How singular that steam should have been among the motive agents of the most ancient idol-worship of Egypt, and that it should formerly have been employed with tremendous effect to delude men and lock them in ignorance, while it now contributes so largely to enlighten and benefit mankind! The instances of the early application of steam make us regret that detailed descriptions of the various and ingenious devices have not been preserved; for while we condemn the contrivers of such as were used for the purpose of delusion, we cannot but admire the ingenuity which they displayed in exhibiting before a barbarous people their gods in the most imposing manner, and with such terrific effect.

The mechanical properties of vapor are similar to those of gases in general. The property which is most important to be considered, in the case of steam, is the elastic pressure. When a vapor or gas is contained in a close vessel, the inner surface of the vessel will sustain a pressure arising from the elasticity of the fluid.

This pressure is produced by the mutual repulsion of the particles, which gives them a tendency to fly asunder, and causes the mass of the fluid to exert a force tending to burst any vessel within which it is confined. This pressure is uniformly diffused over every part of the surface of the vessel in which such a fluid is contained: it is to this quality that all the mechanical power of steam is due.

Steam might be said to be the result of a combination of water with a certain amount of heat, and the expansive force of steam arises from the absence of cohesion between and among the particles of water.

Heat universally expands all matter within its influence, whether solid or fluid. But in a solid body it has the cohesion of the particles to overcome; and this so circumscribes its effect, that in cast-iron, for instance, a rate of temperature above the freezing-point sufficient to melt it causes an extension of only about one-eighth of an inch in a foot. With water, however, a temperature of 212°, or 180° above the freezing-point (and which is far from a red heat), converts it into steam of 1700 times its original bulk or volume.

Steam cannot mix with air while its pressure exceeds that of the atmosphere; and it is this property, with that which makes the condition of a body dependent on its temperature, that explains the condensing property of steam.

In a cylinder once filled with steam of a pressure of 15 pounds or more to the square inch, all air is excluded; now, as the existence of the steam depends on its temperature, by abstracting that temperature (which may be done by immersing the cylinder in cold water or cold air), the contained steam assumes the state due to the reduced temperature, and this state will be water.

The latent or concealed heat of steam is one of the most noteworthy properties. The latent heat of steam, though showing no effect on the thermometer, may be as easily known as the sensible or perceivable heat.

To show this property of steam by experiment, place an indefinite amount of water in a closed vessel, and let a pipe, proceeding from its upper part, communicate with another vessel, which should be open, and, for convenience of illustration, shall contain just 5½ pounds of water at 32°, or just freezing. The pipe from the closed vessel must reach nearly to the bottom of the open one. By boiling the water contained in the first vessel until steam

enough has passed through the pipe to raise the water in the open vessel to the boiling-point (212° Fah.), we shall find the weight of the water contained by the latter to be 6½ pounds. Now, this addition of one pound to its weight has resulted solely from the admission of steam to it; and this pound of steam, therefore, retaining its own temperature of 212°, has raised 5½ pounds of water 180°, or an equivalent to 990°, and, including its own temperature, we have 1201°, which it must have possessed at first.

The sum of the latent and sensible heat of steam is in all cases nearly constant, and does not vary much from 1200°.

The elasticity of steam increases with an increase in the temperature applied, but not in the same ratio. If steam is generated from water at a temperature which gives it the same pressure as the atmosphere, an additional temperature of 38° will give it the pressure of two atmospheres; a still further addition of 42° gives it the tension of four atmospheres; and with each successive addition of temperature of between 40° and 50° the pressure becomes doubled.

An established relation must exist between the temperature and elasticity of steam; in other words, water at 212° Fah. must be under the pressure of the steam naturally resulting from that temperature, and so at any other temperature.

If this natural pressure on the surface of the water be removed without a corresponding reduction in the temperature, a violent ebullition of the water is the immediate result.

Another result attending formation of steam is, that when an engine is in operation and working off a proper supply of steam, the water level in the boiler artificially rises, and shows by the gauge-cocks a supply greater than that which really exists.

As the pressure of steam is increased, the sensible heat is augmented, and the latent heat undergoes a corresponding diminution, and *vice versa.* The sum of the sensible and latent heat is, in fact, a constant quantity; the one being always increased at the expense of the other.

It has been shown that in converting water at 32° of temperature, and under a pressure of 15 pounds per square inch, it was necessary first to give it 180° additional sensible heat, and afterwards 990° of latent heat, the total heat imparted to it being 1170°. Such, then, is the actual quantity of heat which must be imparted to ice-cold water to convert it into steam. The actual temperature to which water would be raised by the heat necessary to evaporate it, if its evaporation could be prevented by confining it in a close vessel, will be found by adding 32° to 1170°.

It may, therefore, be stated that the heat necessary for the evaporation of ice-cold water is as much as would raise it to the temperature of 1202°, if its evaporation were prevented.

If the temperature of red-hot iron be, as it is supposed, 800° or 900°, and that all bodies become incandescent at the same temperature, it follows that to evaporate water it is necessary to impart to it 400° more heat than would be sufficient to render it red-hot, if its evaporation were prevented.

It has been asserted, in some scientific works, that by mere mechanical compression, steam will be converted into water. This is, however, an error; since steam, in whatever state it may exist, must possess at least 212° of heat; and as this quantity of heat is sufficient to maintain it in the vaporous form under whatever pressure it may be placed, it is clear that no compression or increase of pressure can diminish the actual quantity of heat con-

tained in the steam, and it cannot, therefore, convert any portion of the steam into power.

Steam, by mechanical pressure, if forced into a diminished volume, will undergo an augmentation both of temperature and pressure, the increase of temperature being greater than the diminution of volume; in fact, any change of volume which it undergoes will be attended with the change of temperature and pressure indicated in the table on pages 39–43.

The steam, after its volume has been changed, will assume exactly the pressure and temperature which it would have in the same volume if it were immediately evolved from water.

Let us suppose a cubic inch of water converted into steam under a pressure of 15 pounds per square inch, and the temperature of 212°. Then let its volume be reduced by compression in the proportion of 1700 to 930. When so reduced, its pressure will be found to have risen from 15 pounds per square inch to 29½ pounds per square inch; but this is exactly the state as to pressure, temperature, and density the steam would be in if it were immediately raised from water under the pressure of 29½ pounds per square inch. It appears, therefore, that in whatever manner, after evaporation, the density of steam be changed, whether by expansion or contraction, it will still remain the same as if it were immediately raised from water in its actual state.

The circumstance which has given rise to the erroneous notion that mere mechanical compression will produce a condensation of steam, is that the vessel in which steam is contained must necessarily have the same temperature as the steam itself.

Water while passing into steam suffers a great enlargement of volume; steam, on the other hand, in being con-

verted into water, undergoes a corresponding diminution of volume. It has been seen that a cubic inch of water, evaporated at the temperature of 212°, swells into 1700 cubic inches of steam. It follows, therefore, that if a closed vessel, containing 1700 cubic inches of steam, be exposed to cold sufficient to take from the steam all its latent heat, the steam will be reconverted into water, and will shrink into its original dimensions, and will leave the remainder of the vessel a vacuum.

This property of steam has supplied the means, in practical mechanics, of obtaining that amount of mechanical power which the properties of the atmosphere confer upon a vacuum.

The temperature and pressure of steam produced by immediate evaporation, when it has received no heat save that which it takes from the water, have a fixed relation one to the other.

If this relation was known and expressed by a mathematical formula, the temperature might always be inferred from the pressure, and *vice versa.*

But physical science has not yet supplied any principle by which such a formula can be deduced from any known properties of liquids.

The same difficulty which attends the establishment of a general formula expressing the relation between the temperatures and pressures of steam, also attends the determination of one expressing the relation between the pressure and augmented volume into which water expands by evaporation.

In the preceding observations, steam has been considered as receiving no heat except that which it takes from the water during the process of evaporation; the amount of heat of which, as has been shown, is 1170° more than the heat contained in ice-cold water. But steam, after having

been formed from water by evaporation, may, like all other material substances, receive an accession of heat from any external source, and its temperature may therefore be elevated.

If the steam to which such additional heat is imparted be so confined as to be incapable of enlarging its dimensions, the effect produced upon it by the increase of temperature will be an increase of pressure.

But if, on the other hand, it be confined under a given pressure, with power to enlarge its volume, subject to the preservation of that pressure, as would be the case if it were contained in a cylinder under a movable piston loaded with a given pressure, then the effect of the augmented temperature will be, not an increase of pressure, but an increase of volume; and the increase of volume, in this latter case, will be in exactly the same proportion as the increase of pressure in the former case.

These effects of elevated temperature are common, not only to the vapors of all liquids, but also to all permanent gases; but, what is much more remarkable, the numerical amount of the augmentation of pressure or volume produced by a given increase of temperature is the same for all vapors and gases. If the pressure which any gas or vapor would have, were it reduced to the temperature of melting ice, be expressed by 100,000, the pressure which it will receive for every degree of temperature by which it is raised will be expressed by 208⅓, or what amounts to the same, the additional pressure produced by each degree of temperature will be the 480th part of its pressure at the temperature of melting ice.

Steam which thus receives additional heat after its separation from the water from which it is evolved has been called *superheated steam*, to distinguish it from *common steam*, which is that usually employed in steam-engines.

Steam of atmospheric pressure occupies 1669 times the volume of the water from which it is raised, and as a cubic foot of water weighs 62·4 pounds, a cubic foot of steam of atmospheric pressure weighs about ·038 pound. In order to exert a pressure by its mere dead weight of 14·7 pounds per square inch, such steam of uniform density would have to stand at a height of 10½ miles—the velocity due to a fall from this height in 1888 feet per second, and this, accordingly, is the velocity with which steam of atmospheric pressure enters a vacuum. And if the velocity of steam were inversely as its pressure, this would be the velocity of steam of every pressure in moving into a vacuum, since, so far as generating effluent velocity is concerned, the mere elasticity of a gas is inoperative.

The effluent velocity of steam into the atmosphere or into steam of lower pressure, then, has to be carefully considered in the treatment of steam-engines. In the following table, the pressure given in pounds above the atmosphere is 0·3 pound less than the pressure employed in making the calculation:

Pressure above the Atmosphere.	Velocity of Escape per Second.	Pressure above the Atmosphere.	Velocity of Escape per Second.
Pounds.	Feet.	Pounds.	Feet.
1	540	50	1,736
2	698	60	1,777
3	814	70	1,810
4	905	80	1,835
5	981	90	1,857
10	1,232	100	1,875
20	1,476	110	1,889
30	1,601	120	1,900
40	1,681	130	1,909

To saturated steam, or steam as it rises from the water

from which it is generated, these calculations of course only apply. Whatever may be the pressure per square inch common to different conditions of steam, the effluent velocity will be inversely as the square root of the specific gravity of steam. If the steam be superheated, its specific gravity for a given pressure will be diminished, and its velocity of escape into the air or into a vacuum will be increased. If, on the contrary, the steam carry with it any suspended moisture, its specific gravity for a given pressure will be increased and its velocity of escape diminished.

A very important question will probably arise in the mind of the reader as to the amount of work that a given weight of steam is capable of performing. A pound of steam, of say 120 pounds pressure above that of the atmosphere, is virtually a pound of water heated 1681 degrees above the absolute zero of a perfect gas thermometer, 1220° above Fahrenheit's zero, 1188 degrees above the freezing-point, or 1118° above the sensible temperature of steam of one pound absolute pressure per square inch, the lowest pressure at which a condensing-engine could be expected to work. Either of these total temperatures, multiplied by 772, will give the energy in foot-pounds theoretically due to the steam when worked down, say, into water of the corresponding temperature.

But it must be remembered that, as a gas (to which steam in this case is necessarily compared) it would, upon the accepted law of expansion, only lose its elasticity at a temperature below the freezing-point. If we work down to 102°, the temperature of water from which steam of one pound total pressure would escape, we shall have an energy, for one pound weight of steam, of 863,096 foot-pounds; and if 10 pounds of steam be evaporated from 102° by one pound of coal, giving 8,630,960 foot-pounds per pound of coal, an engine working up to the full

power of the steam would require but $\frac{1980000}{8630960} = 0{\cdot}23$ pound, or less than *four ounces* of coal per indicated horse-power per hour, an hourly horse-power being $33{,}000 \times 60 = 1{,}980{,}000$ foot-pounds.

To obtain such a result, the steam must in the very act of doing work be reduced to one pound of water at 102°. This, however, is quite a theoretical deduction, and nothing like it could, with our present knowledge, be approached in practice; especially, as in expanding, the steam is constantly losing heat and liquefying in the very act of doing work, and thus losing pressure apart from the loss due to the apparent enlargement of volume.

Superheated steam admits of losing a part of its heat without suffering partial condensation; but *common* steam is always partially condensed, if any portion of heat be withdrawn from it. But it must be remembered, any additional arrangements for heating the steam can but complicate the machinery, and thus require increase of space, besides adding to the cost of the engine. But these objections are more serious in the case of the marine engine, the boilers of which are mostly fed with sea-water, strongly impregnated with various salts, and particularly with chloride of sodium. At the usual temperature of the steam used for working these engines, which is generally from 250° to 270°, the presence of this salt causes no inconvenience; but when the steam is superheated, chemical decomposition ensues, and the chlorine thus set free attacks all the brass work of the engine with which it comes in contact, and the valves and valve-seats are speedily destroyed and the engine put out of order.

But there can be no doubt whatever but that the use of superheated steam is more economical than that of ordinary saturated steam. In some of the scientific reports on this subject, it has been shown that there is a saving of

from 20 to 25 per cent. in the fuel consumed. This fact has induced inventors to turn their attention to the task of devising some practical appliances for producing steam in this superheated condition.

Motion of Steam.—Steam, if unimpeded, moves with great velocity from one enclosure to another, under very slight differences of pressure. The laws which regulate this movement, though apparently of a simple character, are not so easily reduced to exact formula as would seem desirable. All the rules, therefore, which are given, must be taken with due reserve and with important qualifications. The conditions of the free motion of steam will be exhibited as nearly as science has been able to estimate them.

These conditions are three: Steam may flow into a vacuum, or into the atmosphere, or into steam of less density. The conditions of its flow in all these cases are of course entirely different. In the middle case—that of its flow into the atmosphere—about 15 pounds of its total pressure go for nothing, being expended in overcoming the atmospheric resistance, and before the slightest motion of its own or impulse to any other body is possible. The law applicable to non-elastic fluids is the same as that which applies to gases and steam.

Volume and Weight of Steam.—Seventy-five cubic feet of steam at a pressure of 140 pounds per square inch weigh 26 pounds.

Five cubic feet of steam at a pressure of 75 pounds per square inch weigh 1 pound.

One cubic foot of steam at a pressure of 15 pounds per square inch weighs .038 pound.

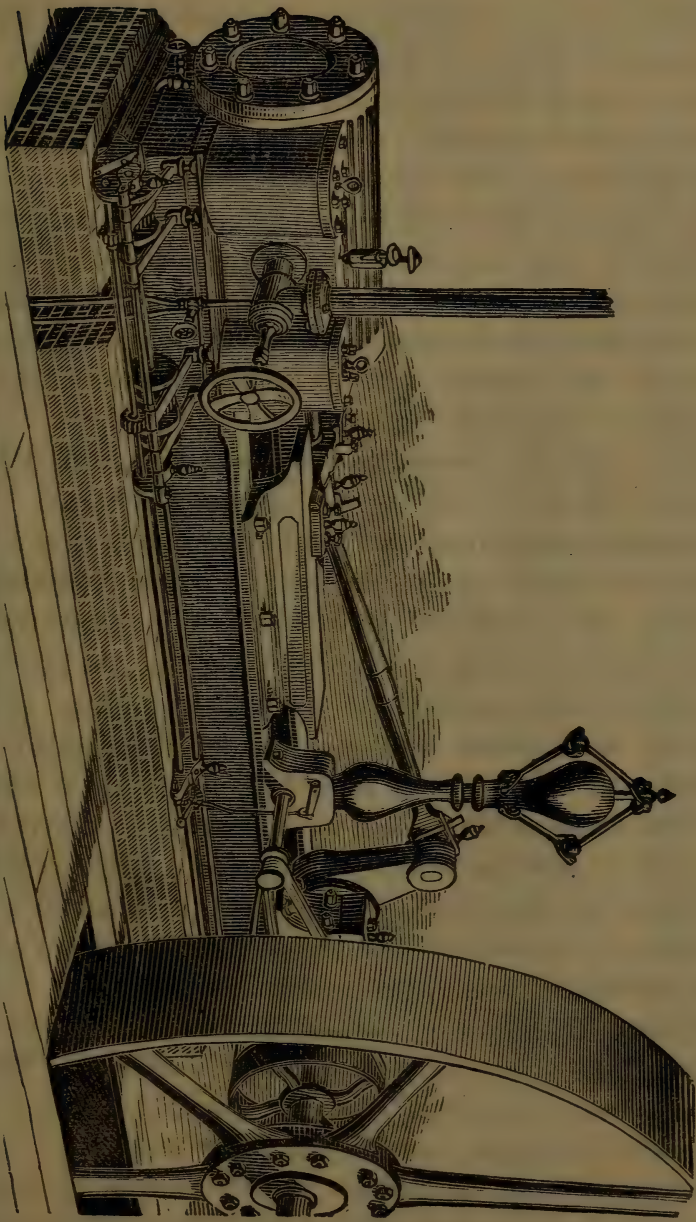
Steam, at any given pressure, always stands at a certain temperature, which is termed the "temperature due to the pressure." Steam follows very nearly the same law that all other gaseous bodies are subject to in acquiring ad-

ditional degrees of heat. The law is, briefly, as follows: That all gaseous bodies expand equally for equal additions of temperature; and that the progressive rate of expansion is equal for equal increments of temperature.

If two volumes of steam of the same weight be compared, we institute a comparison between their relative volumes; for, being of the same weight, they are produced from the same quantity of water. The relative volume of steam being the absolute volume divided by the volume of water from which it was produced, the ratio of any two relative volumes of steam is the same as the ratio of their absolute volumes. So also with steam held in contact with the water in the boiler, the same pressure exhibited by the gauge corresponds to the same temperature in the boiler, and the same temperature in the boiler will always give the same corresponding pressure of steam. Therefore, if we increase the temperature, we increase the pressure and density, and we, of course, get the greatest pressure and density that steam can have at that temperature.

The table on page 39 shows that the saving of fuel is in proportion to the increase of pressure — the advantage of generating and using high-pressure steam is thereby made apparent. The table also shows that the last 10 pounds of additional pressure only require four degrees of heat to raise it; whereas, the first 10 pounds of pressure above the atmosphere require 29 additional degrees of heat to raise it — a difference of 25 degrees.

It also shows that at 212° the total heat of steam is 1178·1°, which gives a difference of 966·1°. This heat, usually termed latent, is absorbed in performing the work of expanding the particles of water from the solid to the gaseous state. Now, suppose the water is evaporated at 60 pounds pressure, the steam will have a temperature of 307°, and a total heat of 1207°. If the feed has been



WATT AND CAMPBELL'S HIGH-PRESSURE CUT-OFF ENGINE.

introduced at 60°, it is evident that 1147° of heat have been imparted.

As the amount evaporated is inversely proportional to the quantity of heat required, we have 1147 ÷ 966 = 1·2. Multiplying by this factor, the quantity evaporated at 60 pounds pressure from 60°, we obtain the amount that would be evaporated at 212° by the same quantity of fuel.

By the same table will be seen the comparatively small increase of heat required to evaporate water at higher pressures. Suppose we take water evaporated at 45 pounds pressure from a feed temperature of 60°, then each pound of water will require 1202·7° — 60 = 1142·7° for its conversion into steam.

If we take the pressure at 100 pounds, we shall have 1216·5 — 60 = 1156·5° as the quantity required. The difference between these two total quantities is only 13·8°, and is so small as to be scarcely worth considering. Leaving out of account the loss due to the slight reduction of the conducting power of the material, the increased amount of heat required for the higher pressure will be only $\frac{1}{80}$ of the total heat required at 60 pounds. The economy of using steam of a high pressure is clearly manifest when, at the same time, advantage is taken of the facilities it offers for working expansively in the cylinder.

Theory has long since demonstrated the economical advantages to be derived from the use of high steam pressures combined with high grades of expansion in the cylinder. The practical difficulties that stood in the way having been gradually and successfully overcome, the result has been the marked changes from the 7 pound and 10 pound pressures, so common forty years ago, to the pressures of from 80 pounds to 100 pounds, at present employed; and the more general employment of the higher pressures will be demanded as the advantages of using steam expansively become more generally recognized.

TABLE

SHOWING THE TEMPERATURE AND WEIGHT OF STEAM AT DIFFERENT PRESSURES FROM 1 POUND PER SQUARE INCH TO 300 POUNDS, AND THE QUANTITY OF STEAM PRODUCED FROM 1 CUBIC INCH OF WATER, ACCORDING TO PRESSURE.

Total pressure per square inch measured from a vacuum.	Pressure above atmosphere.	Sensible temperature in Fahrenheit degrees.	Total heat in degrees from zero of Fahrenheit.	Weight of one cubic foot of steam.	Relative volume of steam compared with water from which it was raised.
1	......	102.1	1144.5	.0030	20582
2	......	126.3	1151.7	.0058	10721
3	......	141.6	1156.6	.0085	7322
4	......	153.1	1160.1	.0112	5583
5	......	162.3	1162.9	.0138	4527
6	......	170.2	1165.3	.0163	3813
7	......	176.9	1167.3	.0189	3298
8	......	182.9	1169.2	.0214	2909
9	......	188.3	1170.8	.0239	2604
10	......	193.3	1172.3	.0264	2358
11	......	197.8	1173.7	.0289	2157
12	......	202.0	1175.0	.0314	1986
13	......	205.9	1176.2	.0338	1842
14	......	209.6	1177.3	.0362	1720
14.7	0	212.0	1178.1	.0380	1642
15	.3	213.1	1178.4	.0387	1610
16	1.3	216.3	1179.4	.0411	1515
17	2.3	219.6	1180.3	.0435	1431
18	3.3	222.4	1181.2	.0459	1357
19	4.3	225.3	1182.1	.0483	1290
20	5.3	228.0	1182.9	.0507	1229
21	6.3	230.6	1183.7	.0531	1174
22	7.3	233.1	1184.5	.0555	1123
23	8.3	235.5	1185.2	.0580	1075
24	9.3	237.8	1185.9	.0601	1036
25	10.3	240.1	1186.6	.0625	996
26	11.3	242.3	1187.3	.0650	958
27	12.3	244.4	1187.8	.0673	926
28	13.3	246.4	1188.4	.0696	895
29	14.3	248.4	1189.1	.0719	866
30	15.3	250.4	1189.8	.0743	838
31	16.3	252.2	1190.4	.0766	813

TABLE—(*Continued*).

Total pressure per square inch measured from a vacuum.	Pressure above atmosphere.	Sensible temperature in Fahrenheit degrees.	Total heat in degrees from zero of Fahrenheit.	Weight of one cubic foot of steam.	Relative volume of steam compared with water from which it was raised.
32	17.3	254.1	1190.9	.0789	789
33	18.3	255.9	1191.5	.0812	767
34	19.3	257.6	1192.0	.0835	746
35	20.3	259.3	1192.5	.0858	726
36	21.3	260.9	1193.0	.0881	707
37	22.3	262.6	1193.5	.0905	688
38	23.3	264.2	1194.0	.0929	671
39	24.3	265.8	1194.5	.0952	655
40	25.3	267.3	1194.9	.0974	640
41	26.3	268.7	1195.4	.0996	625
42	27.3	270.2	1195.8	.1020	611
43	28.3	271.6	1196.2	.1042	598
44	29.3	273.0	1196.6	.1065	595
45	30.3	274.4	1197.1	.1089	572
46	31.3	275.8	1197.5	.1111	561
47	32.3	277.1	1197.9	.1133	550
48	33.3	278.4	1198.3	.1156	539
49	34.3	279.7	1198.7	.1179	529
50	35.3	281.0	1199.1	.1202	518
51	36.3	282.3	1199.5	.1224	509
52	37.3	283.5	1199.9	.1246	500
53	38.3	284.7	1200.3	.1269	491
54	39.3	285.9	1200.6	.1291	482
55	40.3	287.1	1201.0	.1314	474
56	41.3	288.2	1201.3	.1336	466
57	42.3	289.3	1201.7	.1364	458
58	43.3	290.4	1202.0	.1380	451
59	44.3	291.6	1202.4	.1403	444
60	45.3	292.7	1202.7	.1425	437
61	46.3	293.8	1203.1	.1447	430
62	47.3	294.8	1203.4	.1469	424
63	48.3	295.9	1203.7	.1493	417
64	49.3	296.9	1204.0	.1516	411
65	50.3	298.0	1204.3	.1538	405
66	51.3	299.0	1204.6	.1560	399
67	52.3	300.0	1204.9	.1583	393
68	53.3	300.9	1205.2	.1605	388
69	54.3	301.9	1205.5	.1627	383

TABLE—(*Continued*).

Total pressure per square inch measured from a vacuum.	Pressure above atmosphere.	Sensible temperature in Fahrenheit degrees.	Total heat in degrees from zero of Fahrenheit.	Weight of one cubic foot of steam.	Relative volume of steam compared with water from which it was raised.
70	55.3	302.9	1205.8	.1648	378
71	56.3	303.9	1206.1	.1670	373
72	57.3	304.8	1206.3	.1692	368
73	58.3	305.7	1206.6	.1714	363
74	59.3	306.6	1206.9	.1736	359
75	60.3	307.5	1207.2	.1759	353
76	61.3	308.4	1207.4	.1782	349
77	62.3	309.3	1207.7	.1804	345
78	63.3	310.2	1208.0	.1826	341
79	64.3	311.1	1208.3	.1848	337
80	65.3	312.0	1208.5	.1869	333
81	66.3	312.8	1208.8	.1891	329
82	67.3	313.6	1209.1	.1913	325
83	68.3	314.5	1209.4	.1935	321
84	69.3	315.3	1209.6	.1957	318
85	70.3	316.1	1209.9	.1980	314
86	71.3	316.9	1210.1	.2002	311
87	72.3	317.8	1210.4	.2024	308
88	73.3	318.6	1210.6	.2044	305
89	74.3	319.4	1210.9	.2067	301
90	75.3	320.2	1211.1	.2089	298
91	76.3	321.0	1211.3	.2111	295
92	77.3	321.7	1211.5	.2133	292
93	78.3	322.5	1211.8	.2155	289
94	79.3	323.3	1212.0	.2176	286
95	80.3	324.1	1212.3	.2198	283
96	81.3	324.8	1212.5	.2219	281
97	82.3	325.6	1212.8	.2241	278
98	83.3	326.3	1213.0	.2263	275
99	84.3	327.1	1213.2	.2285	272
100	85.3	327.9	1213.4	.2307	270
101	86.3	328.5	1213.6	.2329	267
102	87.3	329.1	1213.8	.2351	265
103	88.3	329.9	1214.0	.2373	262
104	89.3	330.6	1214.2	.2393	260
105	90.3	331.3	1214.4	.2414	257
106	91.3	331.9	1214.6	.2435	255
107	92.3	332.6	1214.8	.2456	253

TABLE—(*Continued*).

Total pressure per square inch measured from a vacuum.	Pressure above atmosphere.	Sensible temperature in Fahrenheit degrees.	Total heat in degrees from zero of Fahrenheit.	Weight of one cubic foot of steam.	Relative volume of steam compared with water from which it was raised.
108	93.3	333.3	1215.0	.2477	251
109	94.3	334.0	1215.3	.2499	249
110	95.3	334.6	1215.5	.2521	247
111	96.3	335.3	1215.7	.2543	245
112	97.3	336.0	1215.9	.2564	243
113	98.3	336.7	1216.1	.2586	241
114	99.3	337.4	1216.3	.2607	239
115	100.3	338.0	1216.5	.2628	237
116	101.3	338.6	1216.7	.2649	235
117	102.3	339.3	1216.9	.2674	233
118	103.3	339.9	1217.1	.2696	231
119	104.3	340.5	1217.3	.2738	229
120	105.3	341.1	1217.4	.2759	227
121	106.3	341.8	1217.6	.2780	225
122	107.3	342.4	1217.8	.2801	224
123	108.3	343.0	1218.0	.2822	222
124	109.3	343.6	1218.2	.2845	221
125	110.3	344.2	1218.4	.2867	219
126	111.3	344.8	1218.6	.2889	217
127	112.3	345.4	1218.8	.2911	215
128	113.3	346.0	1218.9	.2933	214
129	114.3	346.6	1219.1	.2955	212
130	115.3	347.2	1219.3	.2977	211
131	116.3	347.8	1219.5	.2999	209
132	117.3	348.3	1219.6	.3020	208
133	118.3	348.9	1219.8	.3040	206
134	119.3	349.5	1220.0	.3060	205
135	120.3	350.1	1220.2	.3080	203
136	121.3	350.6	1220.3	.3101	202
137	122.3	351.2	1220.5	.3121	200
138	123.3	351.8	1220.7	.3142	199
139	124.3	352.4	1220.9	.3162	198
140	125.3	352.9	1221.0	.3184	197
141	126.3	353.5	1221.2	.3206	195
142	127.3	354.0	1221.4	.3228	194
143	128.3	354.5	1221.6	.3258	193
144	129.3	355.0	1221.7	.3273	192
145	130.3	355.6	1221.9	.3294	190

TABLE—(*Concluded*).

Total pressure per square inch measured from a vacuum.	Pressure above atmosphere.	Sensible temperature in Fahrenheit degrees.	Total heat in degrees from zero of Fahrenheit.	Weight of one cubic foot of steam.	Relative volume of steam compared with water from which it was raised.
146	131.3	356.1	1222.0	.3315	189
147	132.3	356.7	1222.2	.3336	188
148	133.3	357.2	1222.3	.3357	187
149	134.3	357.8	1222.5	.3377	186
150	135.3	358.3	1222.7	.3397	184
155	140.3	361.0	1223.5	.3500	179
160	145.3	363.4	1224.2	.3607	174
165	150.3	366.0	1224.9	.3714	169
170	155.3	368.2	1225.7	.3821	164
175	160.3	370.8	1226.4	.3928	159
180	165.3	372.9	1227.1	.4035	155
185	170.3	375.3	1227.8	.4142	151
190	175.3	377.5	1228.5	.4250	148
195	180.3	379.7	1229.2	.4357	144
200	185.3	381.7	1229.8	.4464	141
210	195.3	386.0	1231.1	.4668	135
220	205.3	389.9	1232.8	.4872	129
230	215.3	393.8	1233.5	.5072	123
240	225.3	397.5	1234.6	.5270	119
250	235.3	401.1	1235.7	.5471	114
260	245.3	404.5	1236.8	.5670	110
270	255.3	407.9	1237.8	.5871	106
280	265.3	411.2	1238.8	.6070	102
290	275.3	414.4	1239.8	.6268	99
300	285.3	417.5	1240.7	.6469	96

WORKING STEAM EXPANSIVELY.

There are two modes of applying the power of steam to the working cylinders of steam-engines, namely: One, allowing steam to flow from the boiler during the whole length of the stroke; and the other, cutting it off from the boiler when the piston has travelled a determined distance — the great and paramount object of this last arrangement being a saving of fuel.

If steam be applied the full length of the stroke, the average pressure will be as the pressure per square inch upon the piston; but if the steam be cut off at half stroke,—suppose the pressure to be 65 pounds per inch when the pressure of the atmosphere is added,—there will be a mean equivalent, or average pressure, throughout the stroke of 55 pounds per square inch, being only 10 pounds less than the full pressure, or 16 per cent. of a loss in power, though half the former quantity of steam has only been used. This alone effects a saving of 34 per cent. in fuel, and shows the great benefit to be derived from expansion in one cylinder.

If this principle be true, and its truth is undeniable, it is quite evident that the greatest economy will result from extending to their full limit the cylinders of steam-engines, and making them of sufficient capacity for this purpose; though with the high-pressures, with which expansion is most available, they will require to be less than are usually made, to allow the engines to produce the maximum effect.

The expansive property of steam is strictly mechanical, and is a property common to all fluids—air, gas, etc. It simply consists in this—that vapor of a given elastic force will expand to certain limits, and during the process of expansion will act on opposing bodies with a force gradually decreasing, causing a diminution of elastic power in an inverse ratio of the increase of volume, until it has reached the limits of its power, or is counterbalanced by the resistance of a surrounding medium. Thus, steam of any given pressure, expanded to twice its original bulk, will exert only one-half its original power.

If a partial vacuum be formed on one side of a piston, its motion will be continued until the density of the steam on the other side be as low as that of the uncondensed vapor on the vacuum side of the piston. It is clear that

the power which may be obtained by thus impelling a piston will be the average between the highest and the lowest pressure upon the piston. It must also be understood that *it is a saving, and not a gain*, that thus results from expansion; a power being made available which was before lost, by using the steam up to its last impelling force, and not allowing it to escape until the whole of that available force has been expended.

This accounts for some engines using more fuel and steam than others, because the steam is not expanded to its utmost limit, in consequence of the steam not being cut off by the valve soon enough, or that the load on the engine is great, and requires the steam to be longer on the piston before it is cut off. If the load on the engine be such as to allow the steam to be cut off early, and to expand to its full available limits in the cylinder, then the most will have been made of it; the highest pressure in the boiler will have been used upon the piston and down to the lowest point.

Were atmospheric air compressed so as to exert a force of 20 pounds on the square inch, and were the supply to be continued throughout the stroke, an impulse would be given to the piston equal to 20 pounds to the square inch during the whole stroke; but if the air was allowed to expand, the impulse would only be as the average, or 10 pounds. It will be evident that, if in the former case the air was suffered to depart from the cylinder at the same elasticity as that which it entered, we should lose the force which was necessary to compress it to its density; while, by expanding it to its limits, we apply every part of that force.

The main-spring of a watch actuates its machinery in this manner: an increasing effort is required to wind up the spring, and a decreasing impulse is given back to the

machinery. But if, after the spring had partially uncoiled itself, it were then liberated, the force which wound it up to its last impelling point would be totally lost.

So in the steam-engine; if the steam be allowed to escape from the cylinder before its force is expanded to the lowest available pressure, the loss will be in proportion to the amount of the pressure not made available.

A certain quantity of fuel is required to raise steam to a certain elasticity. If that steam be allowed, after having moved the piston, to escape into the atmosphere or condenser without having acted expansively, a portion of the fuel which was consumed to raise the steam up to that point of elasticity will have been lost. In one case, a given bulk of fuel would produce fifty; in the other case, it would produce fifty, added to all the intermediates down to the lowest expansive force.

By this it will be apparent that the advantages arising from expansion increase with the density of the steam. In round numbers, 65 pounds of high-pressure steam will perform more than seven times the duty of 25 pounds of low-pressure steam; a fact greatly in favor of high-pressure steam and expansion.

Expansion is, perhaps, the most extraordinary property of steam. The merit of the discovery is due to HORNBLOWER, who, in 1781, obtained a patent for the invention.

The principle of expanding the steam in the condensing engine is the same as in the non-condensing engine, excepting that the steam which exhausts into the atmosphere cannot expand below 15 pounds per square inch, because the exhaust is open to the pressure of the atmosphere in all cases. The resistance of the atmosphere (15 pounds) must be added to the pressure of steam above atmospheric pressure, when calculating the pressure of the expansion of steam upon the piston.

Example.—Steam at 20 pounds pressure above the atmosphere upon the piston, cut off at one-fourth the stroke, will be 8¾ pounds at the termination of the stroke, as shown by the following calculation: 20 pounds added to 15 pounds, the pressure of the atmosphere, equal 35 pounds. This divided by four gives the quotient 8¾ pounds. Thus, 8¾ pounds is the pressure at the termination of the stroke, or 6¼ pounds below atmospheric pressure.

The tables on pages 49 and 50 show the average pressure of steam upon the piston when cut off at any portion of the stroke, beginning at 25 pounds and advancing in 5 pounds up to 135 pounds per square inch, thereby enabling the engineer to determine, at any given pressure, the amount of expansion requisite for the full power to be obtained, and the saving thereby to be effected. In all cases the pressure of the atmosphere must be added to the pressure of the steam above atmosphere, when reference is made to the table for the average throughout the stroke.

Example.—45 pounds of steam above atmosphere upon piston of a high-pressure engine, cut off at one-fourth of the length of the stroke. The average pressure throughout will be, allowing one pound for friction and back pressure to force out the steam in the cylinder, 19¾ pounds. Thus: 45 pounds of steam cut off at one-fourth the stroke, with 15 pounds added, make 60 pounds. Look for 60 on the top line of the table and ¼ on the side. Trace that ¼ to the figures under 60, and the average will be found to be 35¾ pounds. Take 16 pounds from 35¾ pounds for atmospheric pressure and friction, and there remain 19¾ pounds, the available average pressure on the piston.

Example.—30 pounds cut off at one-third. Add 15 = 45. The average in the table will be 31½; deduct 16 pounds, and there remain 15½ pounds, the available average pressure upon the piston.

Another Example.—15 pounds cut off at half-stroke. Add 15 = 30. The average in the table will be 25¼. Deduct 16 pounds, and 9¼ pounds remain, the available pressure.

In these examples the steam in the cylinder has expanded to atmospheric pressure.

In proportion to the pressure of the steam, the cut-off will have to be varied, if the steam is to be expanded to its full limit in the cylinder of a non-condensing engine; that is, down to 15 pounds, or equal to the pressure of the atmosphere.

Rule for ascertaining the Amount of Benefit to be derived from working Steam expansively.—Divide the length of the stroke by the length of space into which steam is admitted; find in the annexed table the hyperbolic logarithm nearest to that of the quotient, to which add one. The sum is the ratio of gain.

TABLE

OF HYPERBOLIC LOGARITHMS TO BE USED IN CONNECTION WITH THE ABOVE RULE.

No.	Logarithm.	No.	Logarithm.	No.	Logarithm.
1.25	.22314	5.	1.60943	9.	2.19722
1.5	.40546	5.25	1.65822	9.5	2.25129
1.75	.55961	5.5	1.70474	10.	2.30258
2.	.69314	5.75	1.74919	11.	2.39789
2.25	.81093	6.	1.79175	12.	2.48490
2.5	.91629	6.25	1.83258	13.	2.56494
2.75	1.01160	6.5	1.87180	14.	2.63905
3.	1.09861	6.75	1.90954	15.	2.70805
3.25	1.17865	7.	1.94591	16.	2.77258
3.5	1.25276	7.25	1.98100	17.	2.83321
3.75	1.32175	7.5	2.01490	18.	2.89037
4.	1.38629	7.75	2.04769	19.	2.94443
4.25	1.44691	8.	2.07944	20.	2.99573
4.5	1.50507	8.5	2.14006	21.	3.04452
4.75	1.55814			22.	3.09104

TABLE

SHOWING THE AVERAGE PRESSURE OF THE STEAM UPON THE PISTON THROUGHOUT THE STROKE, WHEN CUT OFF IN THE CYLINDER FROM $\frac{1}{3}$ TO $\frac{7}{11}$, COMMENCING WITH 25 POUNDS AND ADVANCING IN 5 POUNDS UP TO 75 POUNDS PRESSURE.

Steam cut off in the Cylinder.	Pressure in Pounds at the Commencement of the Stroke.										
	25	30	35	40	45	50	55	60	65	70	75
	Average Pressure in Pounds upon the Piston.										
$\frac{1}{3}$	17½	21	24½	28	31½	35	38½	42	45½	49	52½
$\frac{2}{3}$	23½	28¼	32¾	37½	42	46¾	51½	56¼	61	65½	70¼
$\frac{1}{4}$	15	17¾	20¾	23¾	26¾	29¾	32¾	35¾	38¾	41¾	44¾
$\frac{1}{2}$	21	25¼	29½	33¾	38	42¼	46½	50¾	55	59¼	63½
$\frac{3}{4}$	24	81¾	33½	38½	43¼	48¼	53	57¾	62½	67½	72¼
$\frac{1}{5}$	13	15½	18¼	20¾	23½	26	28½	31¼	34	36½	39
$\frac{2}{5}$	19	23	26¾	30½	39½	38¼	42	46	49¾	53½	57½
$\frac{3}{5}$	22½	26	31½	39¼	40¾	45¼	49¾	54¼	58¾	63¼	67¾
$\frac{4}{5}$	23¾	29¼	34¼	39	44	49	53¾	58½	63½	68½	73¾
$\frac{1}{6}$	11½	14	16¼	18½	20¾	23¼	25½	27¾	30¼	32½	34¾
$\frac{5}{6}$	24½	29½	34½	39½	44¼	49¼	54	59	64	69	73¾
$\frac{1}{7}$	10½	12¼	14¾	16¾	18¾	24	23¼	25¼	27¼	29½	31½
$\frac{2}{7}$	16	19¼	22½	25¾	28¾	32	35¼	38½	41¾	45	48¼
$\frac{3}{7}$	19¾	23¾	27¾	31½	35½	39½	43	47½	51½	55¼	59¼
$\frac{4}{7}$	22¼	26¾	31¼	35½	40	44½	49½	53½	57¾	62½	66¾
$\frac{5}{7}$	23¾	28½	33½	38¼	42¾	47¾	52½	57¼	62	66¾	71½
$\frac{6}{7}$	24¾	29½	34½	39½	44½	49½	54¼	59¼	63¾	69¼	74¼
$\frac{1}{8}$	9½	11½	13½	15¼	17¼	19¼	21¼	23	25	27	28¾
$\frac{3}{8}$	18½	22¼	26	29¾	33½	37	40¾	44½	48¼	52	55¾
$\frac{5}{8}$	22¾	27½	32	36¾	41¼	45½	50½	55¼	59¾	64¼	68¾
$\frac{7}{8}$	24¾	29¾	34¾	39½	44½	49½	54½	59½	64½	69¼	74¼
$\frac{1}{9}$	8¾	10½	12¼	14¼	15¾	17¾	19½	21¼	23	24¾	26½
$\frac{2}{9}$	13¾	16½	19¼	22¼	25	27¾	30½	33¼	36	38¾	41¾
$\frac{4}{9}$	20	24	28	32	36	40¼	44¼	48¼	52¼	56¼	60¼
$\frac{5}{9}$	22	26¼	30¾	35¼	39½	44	48½	52¾	56½	61¾	66
$\frac{7}{9}$	24¼	29	34	38¾	43¾	48½	53½	58¼	63¼	68	72¾
$\frac{8}{9}$	24¾	29¾	34¾	39¾	44½	49½	54½	59½	64½	69½	74½
$\frac{1}{11}$	7¾	9¼	10¾	12¼	13¾	15¼	16¾	18½	20	21½	23
$\frac{2}{11}$	12¼	14¾	17¼	19½	22	24½	27	29½	31¾	34¼	36¾
$\frac{3}{11}$	15½	18¾	21¾	25	28	31¼	34¼	37½	40¾	43¾	47
$\frac{4}{11}$	18¼	21¾	25½	29¼	32¾	36½	40¼	43¾	47½	51	54¾
$\frac{5}{11}$	20¼	24¼	28¼	32½	36½	40½	44½	48¾	52¾	56¾	60¾
$\frac{6}{11}$	21¾	26¼	30½	35	39¼	43¾	48	52½	56¾	61¼	65½
$\frac{7}{11}$	23	27½	32¼	36¾	41½	46	50¾	55¼	60	64½	69¼

TABLE

SHOWING THE AVERAGE PRESSURE OF STEAM UPON THE PISTON THROUGHOUT THE STROKE, WHEN CUT OFF IN THE CYLINDER FROM ⅓ TO 7/9, COMMENCING WITH 80 POUNDS AND ADVANCING IN 5 POUNDS UP TO 130 POUNDS PRESSURE.

Steam cut off in the Cylinder.	Pressure in Pounds at the Commencement of the Stroke.										
	80	85	90	95	100	105	110	115	120	125	130
	Average Pressure in Pounds upon the Piston.										
1/3	56	59½	63	66½	70	73	77½	80½	84	87½	91
2/3	75	79½	84½	89	93¾	98¼	103	107¾	112½	117	121¾
1/4	47¾	50¾	53¾	56¾	59¾	62¾	95½	68½	71½	74½	77½
1/2	67¾	72	76¼	80½	84¾	89	73¼	97¼	101½	105¾	110
3/4	77¼	82	87	91¾	96½	101¼	106¼	111	115¾	120¾	125½
1/5	41¾	44¼	47	49½	52¼	54¾	57¼	60	62½	65¼	67¾
2/5	61¼	65	69	72¾	76½	80¼	84¼	88	91¾	95¾	99½
3/5	72½	77	81½	86	90½	95¼	99½	104¼	108¾	113¼	117¾
4/5	78¼	83	88	92¾	97¾	102¾	107½	112½	117½	122¼	127¼
1/6	37¼	39½	41¾	44¼	46½	48¾	51¼	53½	55¾	58	60¾
5/6	78¾	83¾	88¾	93½	98½	103½	108¼	113¼	118¼	123¼	128
1/7	33½	35¾	37¾	40	42	44	46¼	48¼	50½	52½	54½
2/7	51½	54½	57¾	61	64¼	67½	70¾	74	77¼	80½	83½
3/7	63¼	67¼	71¼	75¼	79	83	87	91	94¾	98¾	102¼
4/7	71¼	75¾	80	84½	89	93½	98	102½	106¾	111½	115¾
5/7	76¼	81	85¾	90¾	95½	100¼	105	109¾	114½	119¼	124
6/7	79	84	89	93¾	98¾	103¾	108¾	113¾	118¾	123½	128½
1/8	30¾	32¾	34½	36½	38½	40½	42¾	44½	46¼	48	50
3/8	59½	63	66¾	70½	74¼	78	81¾	85½	89	92¾	96½
5/8	73½	78	82½	87¼	91¾	96½	101	105½	110¼	114¾	119½
7/8	79¼	84¼	89¼	94¼	99	104	109	114	119	124	128¾
1/9	28¼	30¼	31¾	33¾	35½	37¼	39	40¾	42½	44¼	46
2/9	44½	47¼	55	57¾	55½	58¼	61	63¾	66¾	69½	72¼
4/9	64¼	68¼	72¼	76¼	80½	84½	88½	92½	96½	100½	104½
5/9	70½	74¾	79¼	83¾	88	92½	97	101¼	105¾	110¼	114½
7/9	77¾	82¾	87½	92¼	97¼	102	107	111¾	116¾	121½	126½

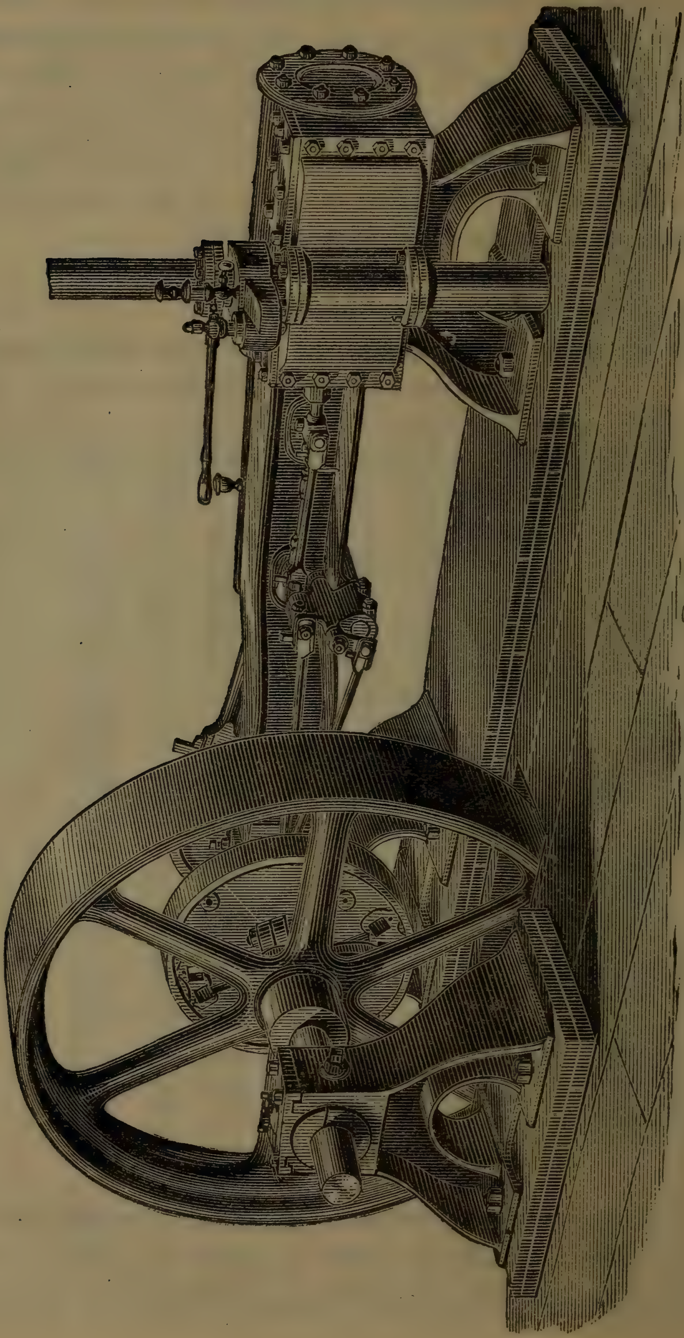
Rule for finding the Mean or Average Pressure in a Cylinder.—Divide the length of the stroke, including the clearance at one end of the cylinder, by the distance, in-

cluding the clearance at one end, that steam follows the piston before being cut off; the quotient will express the relative expansion the steam undergoes. Then find in the following table, in the expansion column, the number corresponding to this, and take the multiplier opposite to it, and multiply the full pressure of the steam per square inch, as it enters the cylinder, by it.

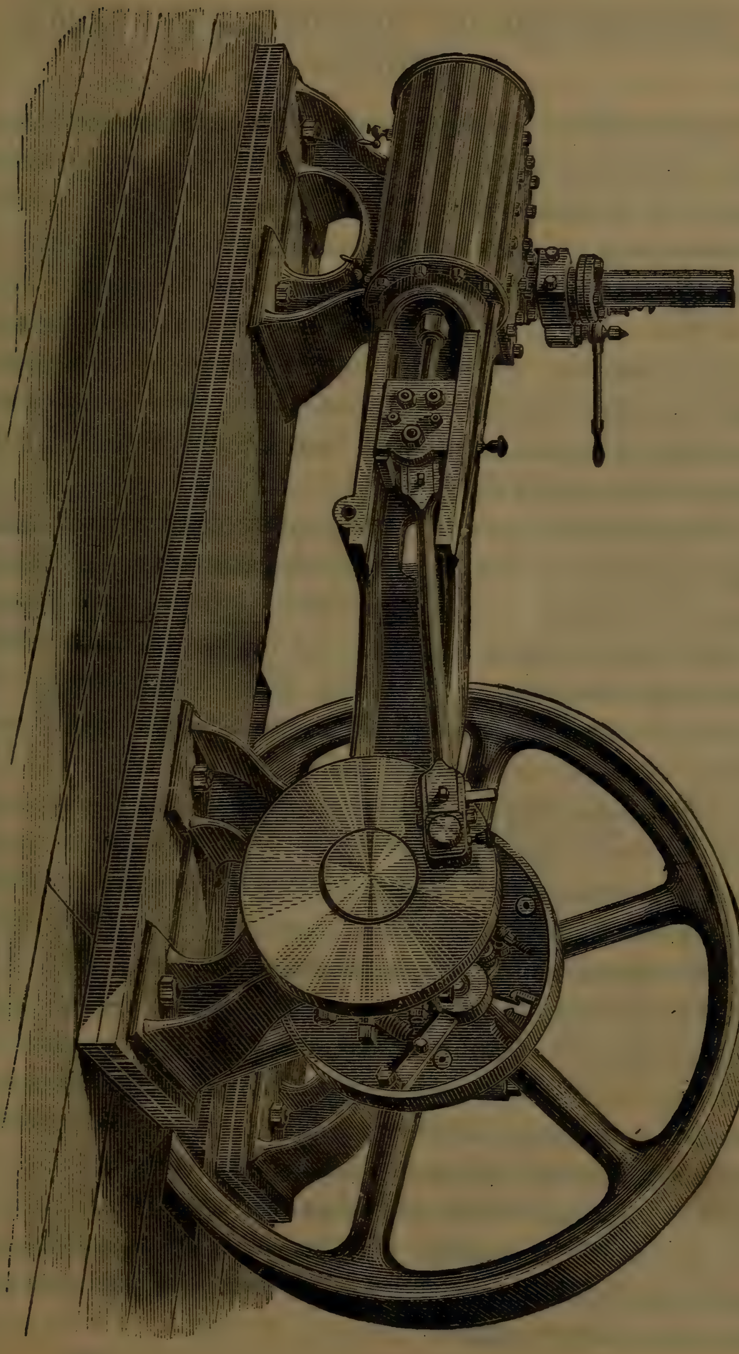
TABLE

OF MULTIPLIERS BY WHICH TO FIND THE MEAN PRESSURE OF STEAM AT VARIOUS POINTS OF CUT-OFF.

Expansion.	Multiplier.	Expansion.	Multiplier.	Expansion.	Multiplier.
1.0	1.000	3.4	.654	5.8	.479
1.1	.995	3.5	.644	5.9	.474
1.2	.985	3.6	.634	6.	.470
1.3	.971	3.7	.624	6.1	.466
1.4	.955	3.8	.615	6.2	.462
1.5	.937	3.9	.605	6.3	.458
1.6	.919	4.	.597	6.4	.454
1.7	.900	4.1	.588	6.5	.450
1.8	.882	4.2	.580	6.6	.446
1.9	.864	4.3	.572	6.7	.442
2.	.847	4.4	.564	6.8	.438
2.1	.830	4.5	.556	6.9	.434
2.2	.813	4.6	.549	7.	.430
2.3	.797	4.7	.542	7.1	.427
2.4	.781	4.8	.535	7.2	.423
2.5	.766	4.9	.528	7.3	.420
2.6	.752	5.	.522	7.4	.417
2.7	.738	5.1	.516	7.5	.414
2.8	.725	5.2	.510	7.6	.411
2.9	.712	5.3	.504	7.7	.408
3.	.700	5.4	.499	7.8	.405
3.1	.688	5.5	.494	7.9	.402
3.2	.676	5.6	.489	8.	.399
3.3	.665	5.7	.484		



BUCKEYE HIGH-PRESSURE CUT-OFF ENGINE.



BUCKEYE HIGH-PRESSURE CUT-OFF ENGINE.

HIGH-PRESSURE OR NON-CONDENSING STEAM-ENGINES.

High-pressure, or non-condensing engines are those engines in which the steam, after its action on the piston, is permitted to escape into the atmosphere, and in which, therefore, the pressure of the outgoing steam must exceed the atmospheric pressure of 15 pounds to the square inch.

In this class of engines are included all locomotive, fire, and nearly all stationary and river-boat engines, which, in turn, comprises a great variety of arrangements and designs known as vertical, beam, inclined, oscillating, trunk, horizontal, etc.

If steam at 30 pounds to the square inch above atmospheric pressure, that is to say, 30 pounds on the steam-gauge, be applied to the piston of a high-pressure engine, it will exert a force equal to the pressure in the boiler above the atmosphere, providing there be sufficient room for the steam, and no obstacle to impede its free flow or lessen its pressure between the boiler and the cylinder; the other side of the piston being open to the atmosphere, and the steam having to overcome the atmospheric pressure in its escape from the cylinder, 15 pounds from the total pressure of 45 pounds will be lost.

Advantages of the High-pressure Engine.—The principal advantages of the high-pressure engine are, its lightness, moderate first cost, economy of space, and the facilities it affords for an increase of pressure and speed, should it become necessary; hence, the high-pressure engine being lighter, more simple, compact, and less expensive in construction, and also less complicated, requiring less skill to manage and less cost to repair, is more desirable for stationary, land, and river-boat purposes than the low-pressure engine.

The high-pressure engine is also desirable in marine steamers on account of economy of room, weight, etc., though objectionable in consequence of its greater consumption of fuel. The causes which occasion this extra consumption of fuel are, first, the steam lost in overcoming the pressure of the atmosphere; second, the loss of heat by radiation in consequence of high pressures and high temperatures; third, the loss occasioned by the escape of heat through the chimney.

In the high-pressure engine, pressure and speed can be increased to any limit within the bounds of safety. Not so, however, in the case of the low-pressure, as, with extremely high pressures and correspondingly high temperatures, it would be impossible to condense the steam, and the result would be a loss of power, occasioned by back pressure resulting from an imperfect vacuum.

For all steam-engines with cylinders less than 24 inches in diameter, the simple high-pressure or non-condensing engine is the most convenient and economical.

POWER OF THE STEAM-ENGINE.

The power which a steam-engine can furnish is generally expressed in "horse-power." It will, therefore, be of interest to engineers, and of special value to many, to have briefly stated what is meant by a "horse-power," and how it has happened that the power of a steam-engine is thus expressed in reference to that of horses.

Prior to the introduction of the steam-engine, horses were very generally used to furnish power to perform various kinds of work, and especially the work of pumping water out of mines, raising coal, etc. For such purposes, several horses working together were required. Thus, to work the pumps of a certain mine, five, six, seven, or

even twenty-five horses were found necessary. When it was proposed to substitute the new power of steam, the proposal naturally took the form of furnishing a steam-engine capable of doing the work of the number of horses used at the same time. Hence, naturally followed the usage of stating the number of horses which a particular engine was equal to, that is, its "horse-power."

But as the two powers were only alike in their equal capacity to do the same work, it became necessary to refer in both powers to some work of a similar character which could be made the basis of comparison. Of this character was the work of raising a weight perpendicularly.

A certain number of horses could raise a certain weight, as of coal out of a mine, at a certain speed; a steam-engine, of certain dimensions and supply of steam, could raise the same weight at the same speed. Thus, the weight raised at a known speed could be made the common measure of the two powers. To use this common measure it was necessary to know what was the power of one horse in raising a weight at a known speed.

By observation and experiment it was ascertained that, referring to the average of horses, the most advantageous speed for work was at the rate of two-and-a-half miles per hour—that, at that rate, he could work eight hours per day, raising perpendicularly from 100 to 150 pounds. The higher of these weights was taken by Watt, that is, 150 pounds at $2\frac{1}{2}$ miles per hour. But this fact can be expressed in another form: $2\frac{1}{2}$ miles per hour is 220 feet per minute $\left(\frac{2\frac{1}{2} \times 5280}{60} = 220\right)$. So, the power of a horse was taken at 150 pounds, raised perpendicularly, at the rate of 220 feet per minute. This also can be expressed in another form: The same power which will raise 150 pounds 220 feet high each minute, will raise

300 pounds	110 feet	high	each	minute.	
3,000 "	11 "	"	"	"	
33,000 "	1 foot "		"	"	

For in each case the total work done is the same, viz., same number of pounds raised one foot in one minute.

If it is clearly perceived that 33,000 pounds, raised at the rate of one foot high in a minute, is the equivalent of 150 pounds at the rate of 220 feet per minute (or $2\frac{1}{2}$ miles per hour), it will be fully understood how it is that 33,000 pounds, raised at the rate of one foot per minute, expresses the power of one horse, and has been taken as the standard measure of power.

It has thus happened that the mode of designating the power of a steam-engine has been by "horse-power," and that one horse-power, expressed in pounds raised, is a power that raises 33,000 pounds one foot each minute. This unit of power is now universally received. Having a horse-power expressed in pounds raised, it was easy to state the power of a steam-engine in horse-power, which was done in the following manner:

The force with which steam acts is usually expressed in its pressure in pounds on each square inch. The piston of a high-pressure steam-engine is under the action of the pressure of steam from the boiler, on one side of the piston, and of the back action of the pressure due to the discharging steam, on the other side. The difference between the two pressures is the effective pressure on the piston; and the power developed by the motion of the piston, under this pressure, will be according to the number of square inches acted on and the speed per minute with which the piston is assumed to move.

Thus, let the number of square inches in the surface of the piston of a steam-engine be 100, and the *effective* pressure on each square inch be 33 pounds, and the movement of the

piston be at the rate of 200 feet per minute, then the total effective pressure on the piston will be $100 \times 33 = 3300$ pounds, and the movement being 200 feet per minute, the piston will move with a power equal to raising 660,000 pounds one foot high each minute, (as $3300 \times 200 = 660,000$,) and as each 33,000 pounds raised one foot high is one-horse power, and $\frac{660,000}{33,000}$ is 20, then the power of this engine is 20-horse power. If this power is used to do work, a part of it will be expended in overcoming the friction of the parts of the engine and of the machinery through which the power is transmitted to perform the work. The calculation made refers to the total power developed by the movement of the piston under the pressure of steam.

The number of feet moved by the piston each minute is known from the length of stroke of piston in feet, and number of revolutions of engine per minute, there being two strokes of the piston for each revolution of the engine. When these three facts are known, the power of an engine can be readily and accurately ascertained; and it is evident that, without the knowledge of each of the facts, viz., square inches of piston, effective pressure on each square inch, and movement of piston per minute, the power cannot be known.

But circumstances, especially those existing when the condensing-engine was introduced by Watt, led to assumptions as to pressure per square inch and speed of piston which, though true at the time, have long since ceased to be true, and consequently the rules based on such assumptions are entirely inapplicable, and when used must of necessity give false statements.

With regard to how much is understood by a horse-power, there is in this country no question at all. Horses vary in their ability to endure protracted labor, and our standard may be more or less than the average of horses

are able to do; but that is of little importance. So long as the number of horse-power of an engine conveys a definite knowledge of its power, it is of little consequence what relation it sustains to the action of any particular class of animals.

FOREIGN TERMS AND UNITS FOR HORSE-POWER.

Countries.	Terms.	Eng. translation.	Units.	English equivalent.
English.	Horse-power.	Horse-power	550 foot-pounds.	550 foot-pounds.
French.	Force de cheval.	Force-horse.	75 kilogr. metres.	542.47 foot-pounds.
German.	Pferde-krafte.	Horse-force.	513 Fuss-funde.	582.25 foot-pounds.
Swedish.	Hast-kraft.	Horse-force.	600 Skalpund-fot.	542.06 foot-pounds.
Russian.	Syl-lochad.	Force-horse.	550 Fyt-funt.	550 foot-pounds.

The French apply the term force de cheval to a power capable of raising 4·5000 kilogrammes 1 metre high in 1 minute, which is equal to a force capable of raising 32,549 pounds 1 foot high in a minute, which is about $\frac{1}{73}$ less than our unit of measure.

Horse-power.	Force de cheval.	Horse-power.	Force de cheval.
10	10.14	60	60.83
15	15.20	65	65.89
20	20.28	70	70.97
25	25.34	75	76.03
30	30.41	80	81.11
35	35.48	85	86.17
40	40.55	90	91.25
45	45.62	95	96.31
50	50.69	100	101.3856
55	55.75		

In this country, and also in England, it has been usual to assign a certain horse-power for a high-pressure engine of certain dimensions; thus, an engine having a cylinder 10 inches in diameter and 24 inches stroke of piston

would be called a 25-horse-power engine, and so on with high-pressure engines of all dimensions. But it is utterly impossible to say what horse-power an engine of the above dimensions would be, unless we knew the effective pressure to be exerted against the piston, and also the speed at which the piston is intended to move.

There are several kinds of horse-power referred to in connection with the steam-engine,—the "nominal," "indicated," "actual or net," "dynamometrical," and "commercial."

The nominal horse-power is admitted to be a force capable of raising a weight of 33,000 pounds one foot high in one minute, or 150 pounds 220 feet high in the same length of time. The term "*nominal horse-power*," as before stated, originated at the time of the discovery of the steam-engine, from the necessity which then arose for comparing its powers with those of the prevailing motor. The *nominal horse-power* was based on the general principle of the age, which dealt with low pressures and slow piston speeds. These quantities have of late years been greatly increased, and the old formula, in consequence, has become of less importance as a true expression of relative capacity. Hence, the term *nominal horse-power* is in reality of itself nominal, as Watt, in order to have his engines give satisfaction, added some twenty-five per cent. to the real work of the best horses in Cornwall.

But the term *nominal horse-power* implies the ability to do so much work in a certain period of time; and, in order to have a proper idea of it, a unit of measure is also employed. This unit is called a horse-power, and, as before stated, is equal to 33,000 pounds raised through a space of one foot in one minute: it is the execution of 33,000 foot-pounds of work in one minute.

Work is performed when a pressure is exerted upon a

body, and the body is thereby moved through space. The unit of pressure is one pound, the unit of space one foot, and work is measured by a "foot-pound" as a unit. Thus, if a pressure of so many pounds be exerted through a space of so many feet, the number of pounds is multiplied into the number of feet, and the product is the number of foot-pounds of work; hence, if the stroke of a steam-engine be seven feet, and the pressure on each square inch of the piston be 22 pounds, the work done at each single stroke, for each square inch of the piston, will be 7 multiplied by 22, equal to 154 foot-pounds.

Indicated Horse-power.—The indicated horse-power is obtained by multiplying together the mean effective pressure in the cylinder in pounds per square inch, the area of the piston in square inches, and the speed of the piston in feet per minute, and dividing the product by 33,000; and as the effective pressure on the piston is measured by an instrument called the indicator, the power calculated therefrom is called the *indicated* horse-power.

Actual or Net Horse-power.—The actual or net horse-power expresses the total available power of an engine; hence it equals the indicated horse-power less an amount expended in overcoming the friction. The latter has two components, viz., the power required to run the engine, detached from its load, at the normal speed, and that required when it is connected with its load. For instance, if an engine is desired to drive 10 machines, each requiring 10-horse power, it should be of sufficient size to furnish 100 *net horse-power*; but to produce this would require about 115 or 20 indicated horse-power. The net horse-power of an engine may be determined by subtracting from the indicated horse-power the power required to overcome the friction of the engine when in the regular performance of its duty.

Dynamometrical Horse-power.—The dynamometrical horse-power is the net power of the engine after allowing for friction, etc., and this alone is the power with which users of steam-engines are concerned. Though not equal in point of accuracy to the indicator, the dynamometer gives the actual power of small engines near enough for all practical purposes; but it cannot be conveniently applied to large engines.

Commercial Horse-power.—The term commercial horse-power is not generally used, and, when used, has no definite meaning, as there is no recognized standard in use among engineers and manufacturers by which to buy and sell engines. Though the question has often been discussed, and its importance generally recognized, it has never been universally adopted, consequently, the nominal horse-power of a steam-engine means anything that the manufacturer feels disposed to call it. It seems very strange that this should be so, as every civilized country has its standard of weights and measures, with strict laws compelling the observance of these standards in the various operations of trade. The public, also, are keenly alive to the importance of these regulations, and no purchaser is so unmindful of his own interests as not to insist on obtaining the full weight of most articles for which he pays; but steam-engines are almost universally bought and sold by a system of guess-work which would not for a moment be tolerated, were it attempted to be practised in any other branch of trade. There is great need of some recognized standard that would designate the number of square inches in the cylinder, travel of piston in feet per minute, and average steam pressure through the length of stroke, that should constitute the commercial horse-power of engines, say, for instance, 4 square inches in the cylinder, a piston speed of 240 feet per minute, and an

average pressure of 40 pounds per square inch; such proportions would be capable of developing a horse-power in most ordinary high-pressure engines, without the necessity of excessive speed or undue straining.

Small engines are generally more economical than large ones, where the steam pressures, points of cut-off, and power developed are the same; for, although the smaller engine, at the same speed, would be less economical at the higher speed necessary to produce the same power, the gain due to high speed overbalances the loss due to the smaller size of cylinder.

Engines too large for the work to be done are less economical than if proportionate to the power required; for instance, an engine of 40-horse power doing the work of 20-horse power, and running at a high speed, the steam would necessarily have to be throttled down by the governor from, say, 60 or 70 pounds boiler pressure to 25 or 30 pounds on the piston, which would be a loss of nearly $\frac{3}{5}$ in fuel, as the loss by atmospheric pressure in non-condensing engines is equally as much for 25 pounds as for 100 pounds pressure.

The steam necessary to drive a 40-horse power high-pressure engine with *no load*, would give more than 10-horse power in a small engine. The cylinder of any engine should be of sufficient size to give the full power required, leaving a reasonable margin for variation in pressure, and for recuperative power under sudden increase of load, *and no larger.* Large engines doing the work easily, and at a low pressure, are economical only when the speed is reduced in proportion to the work to be done.

There are three conditions which influence the economy of non-condensing steam-engines: *steam pressure, expansion,* and *speed of piston;* for it will be found, on selecting

any particular *horse-power*, that the highest steam pressures and revolutions and shortest points of cut-off are those which show the greatest economy of steam. When these three conditions are all favorable at the same time, the maximum economy is obtained; but when one or more only is favorable, the results are so modified as often to appear contradictory.

Effective Pressure against the Piston.—The character of the connections between the boiler and cylinder, their length, degree of protection, number of bends, shape of valves, etc., must all be considered in forming an estimate of the initial steam pressure in the cylinder; while the effective pressure will depend upon the point at which the steam is cut off, and the freedom with which it exhausts; as it has been fully demonstrated by experience that the effective pressure against the piston in the cylinder of steam-engines, more particularly slide-valve engines, rarely, if ever, exceeds $\frac{2}{3}$ of the boiler pressure, as the free flow of the steam from the boiler to the cylinder is obstructed by the action of the governor and affected by the character of the connection, as before stated, so that in calculating the horse-power of steam-engines, not more than $\frac{2}{3}$ of the boiler pressure should be taken as the effective pressure in the cylinder.

When comparing the relative merits of different engines, it is of more importance to steam users to look at the actual power an engine is capable of exerting, rather than at the stated nominal horse-power or size of cylinder; as it is no uncommon thing with two engines of the same diameter of cylinder and the same general proportions, that one may be capable of developing much more power than the other, even with a less consumption of coal per actual horse-power.

The nominal horse-power of a high-pressure engine, though never very definitely defined, should obviously

hold the same relation to the actual power as that which obtains in the case of condensing engines, so that an engine of a given nominal power may be capable of performing the same work, whether high pressure or condensing. But whether it does or not, the standard of a horse-power serves as a standard of comparison, and its utility as a unit of reference is not impaired, whether it represents the actual power of one horse or three, so long as the standard is universal. The following rule will be found very convenient for those who may have occasion to estimate the horse-power of high-pressure or non-condensing steam-engines, as it is practical and correct.

Rule for finding the Horse-power of Steam-engines. — Multiply the area of the piston by the average steam pressure per square inch; multiply this product by the travel of piston in feet per minute; divide this product by 33,000, and the quotient will be the horse-power.

EXAMPLE I.

Diameter of cylinder in inches.....................	10
	10
Square of diameter of cylinder..........	100
Multiplied by the decimal.....................	.7854
Area of piston.......	78.54 inches.
Boiler pressure, 60 pounds; cut-off, ½ stroke, Average pressure in cylinder, 50 pounds;* 5 off for loss by condensation, etc.,	45 lbs.
	39270
	31416
	3534.30
Travel of piston in feet per minute †	250
Divide by..	33,000)883575.00
	26. horse-pow.

* See Tables of Average Pressures, pages 49, 50.

† *To find the Travel of Piston in Feet per Minute.*—Multiply the distance travelled for one stroke in inches by the whole number of strokes in inches, and divide by 12.

EXAMPLE II.

Diameter of cylinder in inches....................	10
	10
Square of diameter of cylinder....................	100
Multiplied by..	.7854
Area of piston	78.54 inches.
Boiler pressure, 80 pounds; cut-off, ¼ stroke, Average pressure in cylinder, 47¾ pounds; 5 off for loss by condensation, etc.,	42.75 lbs.
	39270
	54978
	15708
	31416
	3357.5850
Travel of piston in feet per minute,	300
Divided by................................	33,000)1007275.5000
	30.* horse-power.

EXAMPLE III.

Diameter of cylinder in inches....................	20
	20
Square of diameter of cylinder....................	400
Multiplied by..	.7854
Area of piston	314.1600
Boiler pressure, 60 pounds; cut-off, ¾ stroke, Average pressure in cylinder, 57 pounds; 5 off for loss by condensation, etc.,	52 lbs.
	6283200
	15708000
	16336.3200
Travel of piston in feet per minute,	300
Divided by................................	33,000)4900896.0000
	148.* horse-power.

* In these examples, the fractional parts of a horse-power have been intentionally left out.

EXAMPLE IV.

Diameter of cylinder in inches.........................	20
	20
Square of diameter of cylinder.........................	400
Multiplied by...	.7854
Area of piston	314.1600 inches.
Boiler pressure, 85 pounds; cut-off, ¼ stroke, Average pressure in cylinder, 50 pounds; 5 off for loss by condensation, etc.,	45 lbs.
	15708000
	12566400
	14137.2000
Travel of piston in feet per minute.............	350
	7068600000
	424116000
Divided by....................................	33,000)4948020.0000
	149. horse-power.

It will be seen from the foregoing examples, that any increase of pressure and piston speed makes a very noticeable difference in the power of the engine; but this augmentation of power is not obtained without an increased quantity of steam in proportion to the increased pressure and speed, except where the steam is expanded to its lowest available limits.

A high-pressure engine, for instance, working with 40 pounds steam above the atmospheric pressure upon the piston, cut off at one-third, and expanding the remainder of the stroke, the piston travelling 220 feet per minute, would only exert the power for which it was nominally calculated, independent of friction; but take the same engine, and increase the speed from 220 to 440 feet per minute,—which is quite practicable,—the power of that engine would then be doubled, less the extra friction; but double the quantity of steam would have been used.

Suppose steam at 80 pounds pressure was introduced to the same cylinder, cut off at one-third and worked expansively, as in the first case, the power given out by the 80 pounds pressure would be less in proportion than that at 40 pounds, as the exhaust would be thrown away at about five times the pressure that it was at 40 pounds, and a portion of the useful effect of the steam would be lost, in consequence of its not being expanded to its full limit, the lost portion escaping into the atmosphere and exerting a corresponding back pressure on the piston.

The following method will be found very convenient, as it somewhat abbreviates the rule used in the foregoing examples for calculating the horse-power of a steam-engine.

TABLE OF FACTORS.

Diameter of Cylinder in Inches.	Factor.	Diameter of Cylinder in Inches.	Factor.
8	.152	26	1.608
10	.238	30	2.142
12	.342	36	3.084
14	.466	40	3.808
16	.609	45	4.82
18	.771	48	5.483
20	.952	50	5.95
22	1.151	56	7.463
24	1.37	60	8.568

Rule.—Multiply the factor of the given diameter of cylinder by the speed of piston in feet per minute (using all below hundreds as decimals); multiply the product by the average pressure in pounds per square inch. This last product will be the horse-power of the engine.

EXAMPLE.

Diameter of cylinder, 12 inches,	.342
	2.40
Travel of piston per minute, 240 feet,	13680
	684
Average pressure, 42 pounds,	.82080
	42
	164160
	328320
	34.47360 horse-power.

Bourne's Rule for ascertaining the nominal horse-power of a high-pressure steam-engine working about four times the usual speed.

Multiply the square of the diameter of the cylinder by the pressure on the piston per square inch less a pound and a half, and by the cube root of the stroke in feet, and divide the product by 235. The quotient is the power of the high-speed engine in nominal horse-power.

EXAMPLE.

Diameter of cylinder,	20
	20
	400 sq. of diameter.
Pressure per sq. in. 80 lbs., less $1\frac{1}{2}$ lbs.,	$78\frac{1}{2}$
Stroke 36 inches = 3 feet,	31400
	1.4422496
Divisor,	235)45286.6374400(192.7 N. H. P.

Colburn's Rule is to multiply the area of the piston, the pressure of steam per square inch, the number of revolutions per minute, and the length of stroke together; divide the product by 33,000, and take $\frac{7}{10}$ of the quotient. But Colburn's rule is not correct, as only one-half the piston speed is employed to get the power of the engine. In fact, neither Colburn's nor Bourne's rules are correct.

WASTE IN THE STEAM-ENGINE.

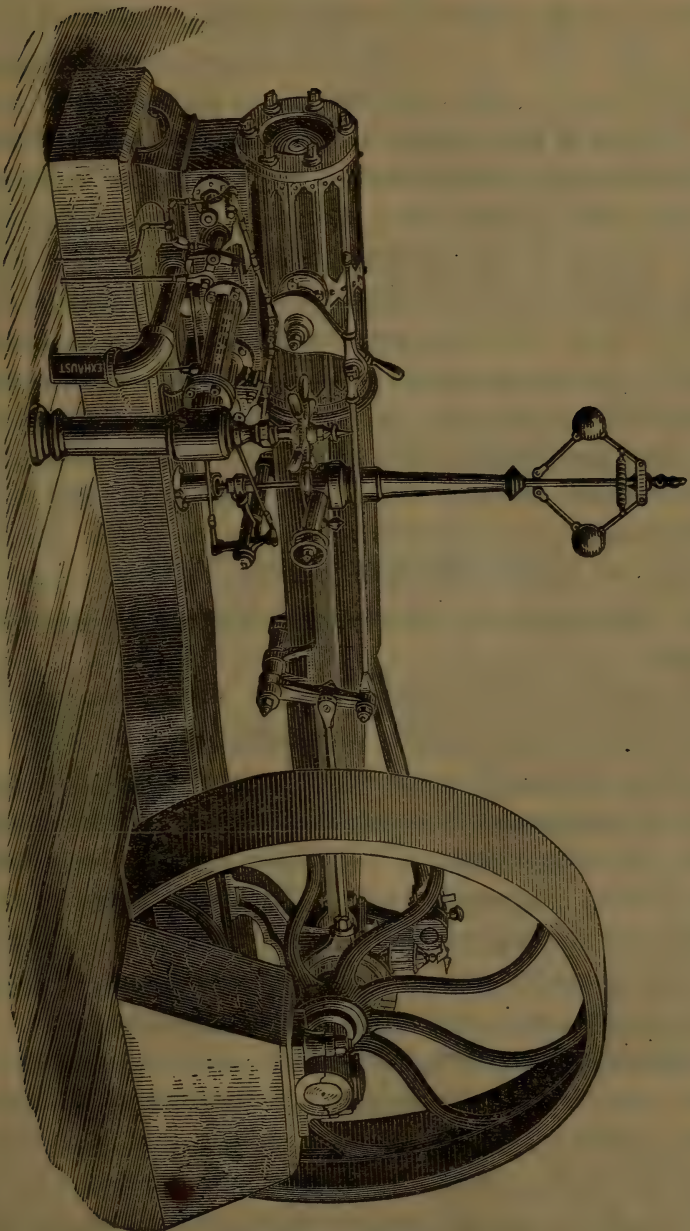
A pound of good coal, it is universally admitted, will liberate, during complete combustion, over 14,500 units of heat, each unit being equivalent to 772 foot-pounds. The mechanical equivalent of the heat developed by the combustion of a pound of coal is, therefore, say $14{,}500 \times 772 = 11{,}000{,}000$ foot-pounds. A horse-power is always assumed to be equal to 33,000 foot-pounds per minute, or 1,980,000 foot-pounds per hour.

So, the combustion of each pound of coal per hour liberates heat enough to develop $11{,}000{,}000 \div 1{,}980{,}000 =$ say 5-horse power; and in a perfect steam-engine the consumption of coal would be about at the rate of one-fifth of a pound per hour for each horse-power developed.

The greatest economy yet obtained in the best high-pressure engines may be taken at from 3 to 4 pounds of coal per indicated horse-power per hour; but for ordinary high-pressure engines in this country and in England a consumption of from 7 to 9 pounds is quite common. In good modern high-pressure steam-engines the useful effect obtained from the work stored up in the fuel may be thus calculated:

Lost through bad firing and incomplete combustion	10 per cent.
Carried off by draft through chimney	30 " "
Carried away in the exhaust steam	50 " "
Utilized in motive power (indicated)	10 " "
	100

The minor causes of loss in the steam-engine are radiation of heat from the boiler, steam-pipes, and cylinder, leakage and condensation; but the great loss arises from the escape of the steam into the atmosphere with only a small portion of its heat utilized; this of itself leads to a loss of from 40 to 60 per cent.; a further loss of useful



WHEELOCK'S HIGH-PRESSURE CUT-OFF ENGINE.

effect in the steam-engine ensues from a portion of the motive power actually developed being absorbed by friction, the useful power of the engine being frequently reduced by this cause by from 10 to 15 per cent.

The use of good material, good workmanship, thorough lubrication and cleanliness, it is true, go far to lessen the friction and increase the efficiency of steam-engines; so also the use of high-pressure steam, high rates of expansion, efficient feed-water heaters, non-conductors and steam-jacketing, is conducive to economy; but what is needed to render the steam-engine what it should be, is complete combustion of the fuel in the furnace, the transfer of all the heat generated to the water in the boiler, the passage of the steam through the engine without the loss of heat, except such as is converted into motive power, the transmission of the remaining heat in the exhaust steam to the feed water, and the absence of friction in its working parts.

In consequence of the enormous waste incurred in the use of the steam-engine, numerous attempts have been made to supersede steam as a prime mover, but as yet without success, as there are certain difficulties connected with the employment of all other agents which have as yet proved insurmountable. In short, there is not at present on the horizon the faintest dawn of the appearance of any mode of generating force calculated to compete with, much less to supersede, the steam-engine.

Electro-magnetism, from which, at one time, so much was expected, is now thoroughly understood to be a far more costly mode of obtaining power than the combustion of coal. Heat, electricity, magnetism, chemical affinity, force, are all equivalent to each other, according to ratios which are fixed and unalterable. The atomic weight of carbon is 6; that of zinc, 32. One pound of carbon

will develop more heat, and consequently more force, than 5 pounds of zinc; whilst, weight for weight, the cost of the former to the latter is as 1 to 50.

The discovery of a new motor, even if such a thing should happen, would take a quarter of a century to replace the present arrangements; and even then it would be the duty of the engineer and the inventor to strive to improve the modes of employing the agent we now possess, and to inquire in which direction further progress in its economical application would lead.

That great improvements can and will be made in the economical working of the steam-engine, none can doubt, who have compared its theoretical capabilities with its present performances.

DESIGN OF STEAM-ENGINES.

The most valuable features of a steam-engine are strength, durability, simplicity, and economy; therefore, in designing an engine, symmetry should be observed, in order that all the working parts may be accessible without any disarrangement of details; lightness should also be adhered to as far as compatible with strength.

The economical working of a steam-engine depends upon many things,—proportions of design, good workmanship, care in using the materials employed in its construction, and a skilful adjustment of the different parts; as the resistance of every machine is increased or diminished according to the harmony of proportions existing between its different principal parts.

The wear of the shaft, the burden on the beam, the wear and tear of the cylinders and packing-rings, the weight borne by the guides in sustaining or directing the cross-head, should all be duly considered by the designing engineer.

Steam-engines embrace a great variety of designs, viz., the vertical, inclined, inverted, beam, horizontal, side-lever, oscillating, trunk, steeple, etc. Those most generally used for stationary engines are the horizontal, vertical, and beam; but there is no other form which stands the tests and meets the public wants as does the horizontal; as an evidence of which, wherever steam-engines of great power are required, the horizontal style of engine is better liked than any other type.

THE BED-PLATE.

The bed, or bed-plate, should be cast in one piece whenever the circumstance of design, construction, etc., will permit. The metal should be so distributed as to give it the necessary resisting strength without excessive weight, as it is not always the heaviest bed-plates that possess the greatest rigidity.

CYLINDERS.

The cylinder being one of the most important and expensive parts of the steam-engine, in order to render it durable and reliable, it should be mathematically correct as to its inside diameter from end to end; though, unfortunately, this is not always the case. But there are several reasons which may be assigned as the cause of irregularities in the bore of steam cylinders, — the machine with which it is bored, the kind of tool used, speed of cutter, want of uniformity in the casting, etc.

Experience has shown it to be advisable, when circumstances will permit, to bore cylinders upright, taking out a heavy cut at first, and then bringing the interior of the cylinder, by successive cuts, to within $\frac{1}{32}$ of an inch of the required size; the remaining portion should then be re-

moved by a cutter which would be neither round nor diamond pointed, but a kind of combination of the two.

It is not at all desirable that a cylinder when bored should present a dead smooth surface, as such surfaces in steam cylinders do not wear so well at the outset as those slightly ridged. This may be accounted for by the extent of surface to be worn down to a steam-tight bearing.

Cylinders should never be removed from the bed-plate for the purpose of reboring, unless it be found impracticable to perform that operation while in its original position; as, if the cylinder be out of line and the crank-pin be in line, the cylinder can be rebored in line with the crank-pin; or if, as is often the case, the crank-pin be out of line with the cylinder, the pin can be removed and the hole in the crank rebored in accurate line with the cylinder. Of course, in such cases, it becomes necessary to have a new crank-pin.

The thickness of steam cylinders cannot be deduced from any fixed rule which would be practical in all cases. For instance, a cylinder 6 inches in diameter should equal at least $\frac{5}{8}$ of an inch in thickness; whereas one 24 inches in diameter would be $1\frac{1}{2}$ inches in thickness: the former equals $\frac{1}{96}$ of the diameter, while the latter equals $\frac{1}{16}$ of the diameter.

TABLE

SHOWING THE PROPER THICKNESS FOR STEAM CYLINDERS OF DIFFERENT DIAMETERS.

Diam. of Cylinder.	Thickness.	Diam. of Cylinder.	Thickness.
6 inch.	$\frac{5}{8}$ inch.	14 inch.	1 inch.
8 "	$\frac{11}{16}$ "	15 "	$1\frac{1}{16}$ "
9 "	$\frac{3}{4}$ "	17 "	$1\frac{1}{8}$ "
10 "	$\frac{13}{16}$ "	18 "	$1\frac{3}{16}$ "
11 "	$\frac{7}{8}$ "	19 "	$1\frac{1}{4}$ "
12 "	$\frac{15}{16}$ "	21 "	$1\frac{3}{8}$ "

The foregoing thicknesses include the proper allowance for reboring. But when the speed of the piston is intended to exceed 300 feet per minute, $\frac{1}{16}$ of an inch should be added per 100 feet to the thickness given.

The following table, however, is more in accordance with modern practice.

Diam. of Cylinder.	Thickness.	Diam. of Cylinder.	Thickness.
6 in.	.440	18 in.	1.070
8 "	.545	20 "	1.175
10 "	.650	22 "	1.280
12 "	.755	24 "	1.385
14 "	.860	26 "	1.490
16 "	.965	28 "	1.595
		30 "	1.700

Rule *for finding the required Diameter of Cylinder for an Engine of any given Horse-power, the Travel of Piston and available Pressure being decided upon.*—Multiply 33,000 by the number of horse-power; multiply the travel of piston in feet per minute by the available pressure in the cylinder. Divide the first product by the second; divide the quotient by the decimal ·7854. The square root of the last quotient will be the required diameter of cylinder.

PISTONS.

The piston ranks next in importance to the cylinder, as the quantity of fuel consumed, the useful effect of the steam, the amount of power developed by the engine, etc., depend in a great measure on the character and condition of the piston. It is, therefore, not to be wondered at, that an immense amount of thought and mechanical skill has been devoted to its improvement, and that its condition is more frequently a source of loss and anxiety to owners of steam-engines, and annoyance to the engineer, than any other detail of the steam-engine.

PISTON-RINGS.

Piston-rings should be of a softer material than that of the cylinder, in order to prevent as far as possible the wear of the latter instead of the rings, as the expense of renewing them is trifling compared with that of reboring the cylinder. Cast-iron is very generally used, as it possesses very many advantages for this purpose, among which are cheapness, durability, uniform expansion, and the additional advantage that it acquires a finer surface and generates less friction than any other material that may be used; although brass rings faced with Babbit-metal are very frequently used for the pistons of locomotives and marine engines.

Piston-rings should be fitted so as to move freely between the flange of the piston-head and the follower-plate, in order that they may accommodate any inequality that may exist in the cylinder; and as their edges are liable to corrode and become leaky, they should be frequently removed from the cylinder and faced-up in a lathe, and reground, and fitted to the flange and follower-plate.

PISTON-SPRINGS.

There is not, in the whole range of steam engineering, it may be safely said, a subject on which the inquiring engineer finds such a dearth of information as he does in regard to the proper amount of elasticity required for the elliptic springs so generally used in adjusting or setting out piston-packing. For, while some bear a slight resemblance to, and possess some of the qualities of, an elliptic spring, others can be said to be nothing more than unshapely pieces of steel.

If any engineer wishes to test the elasticity of such springs, let him place the extreme points of one on two

parallel pieces of iron, and place weights on it, and observe the enormous load it requires to deflect the spring even $\frac{1}{32}$ of an inch in its centre; this will enable him to form some idea of the amount of friction produced in steam cylinders by badly proportioned springs. Such springs, when pressed against the packing-rings by means of set-screws, are as rigid as jack-screws, or solid blocks of iron, and possess no advantages over solid pistons, save that they can be readjusted to take up the wear.

The pistons of large marine engines generally have lighter springs than many small engines, and are not packed so tight by many degrees pressure, in proportion to their areas, as some stationary engines. This may be accounted for by the fact, that pistons that would be perfectly steam-tight under a pressure of 25 to 30 pounds to the square inch, would be apt to leak excessively under a pressure of from 80 to 100 pounds.

STEAM-PISTONS.

The chief merits of the steam-piston seem to consist in its diminished friction and first cost, as it can be more cheaply constructed than spring-pistons, and, after being once put in the cylinder, requires no further attention or adjustment on the part of the engineer; but it is claimed, by its opponents, that it is liable to leak, and wear the cylinder out of "true," in consequence of its being influenced by varying steam pressures. Nevertheless, the steam-piston, after encountering a good deal of prejudice, like many other innovations in steam engineering, is fast establishing its merits with engineers and steam users.

SOLID PISTONS.

Solid pistons are sometimes used, and, when well designed and fitted, answer very well, as they have the ad-

vantage of producing no friction; but they are only practicable in cases where the cylinder is of uniform bore all through, and the engine is perfectly in line. They have the disadvantage of not being adjustable when they become worn or leaky; although there is one solid piston, "Bucks Patent," that can be adjusted to the cylinder more accurately than any spring-piston.

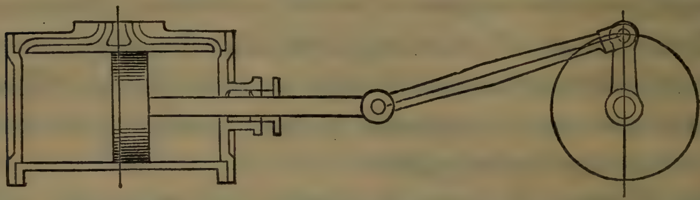
The pistons of steam-hammers are generally made solid, and are kept steam-tight by turning grooves around the head, into which narrow steel or wrought-iron rings are sprung, which adjust themselves to the form of the cylinder, it being the only kind of piston-head capable of resisting the immense jar to which the mass is subjected. The pistons of small steam-engines are frequently made in the same way, as they are found to be very cheap and convenient.

TABLE

OF PISTON SPEEDS FOR ALL CLASSES OF ENGINES — STATIONARY, LOCOMOTIVE, AND MARINE.

Small Stationary Engines,		200 to 250	feet	per minute.
	Average	225	"	"
Large Stationary Engines,		275 to 350	"	"
	Average about	312	"	"
Corliss Engines,		400 to 500	"	"
	Average	450	"	"
Locomotives and Allen Engines,		600 to 800	"	"
	Average	700	"	"
Engines of River Steamers,		400 to 500	"	"
	Average	450	"	"
Engines of Ocean Steamers,		400 to 600	"	"
	Average	500	"	"

PISTON, CONNECTING-ROD, AND CRANK CONNECTION.



The annexed cut shows the position of the piston in the cylinder when the crank is at half stroke. It will be observed that the piston is ahead of its proper position throughout the forward stroke, and that it must of necessity lag behind its position on the return-stroke, and that the points of full power are not on exactly the opposite sides of the diameter of the circle described by the crank, and that a straight line passing through the centre of the crank-shaft cannot intersect both points. These irregularities, which are due to the influence of the crank and connecting-rod, entirely disappear at the end of each stroke.

The crank of an engine moves six times as far while the piston is travelling the first inch of the stroke as while it is making the middle inch, and a little over twice as far while the piston is moving the second inch, and a trifle over $1\frac{1}{2}$ times as far while the piston moves the third inch, and the fourth inch less than $1\frac{1}{2}$ times as far. The crank also travels less when the piston is making the last inch of the stroke than it does while it is making the first.

An explanation of the objections formerly urged against the employment of the crank as a means of converting a reciprocating into a rotary motion, namely, that its leverage, and therefore its power, is so variable, will be found under the head of Cranks, page 109.

TABLE

(By permission, from Auchincloss' "Link and Valve Motions.")

SHOWING THE POSITION OF THE PISTON IN THE CYLINDER AT DIFFERENT CRANK ANGLES, ACCORDING TO THE LENGTH OF CONNECTING-ROD.

(*For Back-action Engines, the words* "Forward" *and* "Return" *must be reversed.*)

Piston Position in Cylinder.	Length of Connecting-Rod 4 to 1 of Stroke.			Length of Connecting-Rod $4\frac{1}{2}$ to 1 of Stroke.			Length of Connecting-Rod 5 to 1 of Stroke.		
	Forward Stroke.	Return Stroke.	Diff.	Forward Stroke.	Return Stroke.	Diff.	Forward Stroke.	Return Stroke.	Diff.
	Deg.	Deg.	Deg.	Deg.	Deg.	Deg.	Deg.	Deg.	Deg.
$0.125 = \frac{1}{8}$	$37\frac{3}{8}$	$46\frac{3}{4}$	$9\frac{3}{8}$	$37\frac{5}{8}$	$46\frac{1}{4}$	$8\frac{5}{8}$	$37\frac{7}{8}$	$45\frac{5}{8}$	$7\frac{3}{4}$
0.2	48	$59\frac{1}{2}$	$11\frac{1}{2}$	$48\frac{1}{2}$	$58\frac{3}{4}$	$10\frac{1}{4}$	$48\frac{3}{4}$	$58\frac{1}{8}$	$9\frac{3}{8}$
$0.25 = \frac{1}{4}$	$54\frac{3}{8}$	$66\frac{3}{4}$	$12\frac{3}{8}$	$54\frac{7}{8}$	66	$11\frac{1}{8}$	$55\frac{3}{8}$	$65\frac{3}{8}$	10
0.3	60	$73\frac{3}{8}$	13	61	$72\frac{5}{8}$	$11\frac{5}{8}$	$61\frac{1}{2}$	72	$10\frac{1}{2}$
$0.333 = \frac{1}{3}$	$64\frac{1}{4}$	$77\frac{5}{8}$	$13\frac{3}{8}$	$64\frac{3}{4}$	$76\frac{7}{8}$	$12\frac{1}{8}$	$65\frac{3}{8}$	$76\frac{1}{4}$	$10\frac{7}{8}$
$0.375 = \frac{3}{8}$	$68\frac{7}{8}$	$82\frac{3}{4}$	$13\frac{7}{8}$	$69\frac{1}{2}$	82	$12\frac{1}{2}$	$70\frac{1}{4}$	$81\frac{3}{8}$	$11\frac{1}{8}$
0.4	$71\frac{3}{4}$	$85\frac{5}{8}$	$13\frac{7}{8}$	$72\frac{3}{8}$	$84\frac{7}{8}$	$12\frac{1}{2}$	73	$84\frac{1}{4}$	$11\frac{1}{4}$
0.45	$77\frac{3}{8}$	$91\frac{1}{2}$	$14\frac{1}{8}$	$78\frac{1}{8}$	$90\frac{5}{8}$	$12\frac{1}{2}$	$78\frac{5}{8}$	90	$11\frac{3}{8}$
$0.5 = \frac{1}{2}$	$82\frac{7}{8}$	$97\frac{1}{8}$	$14\frac{1}{4}$	$83\frac{3}{4}$	$96\frac{1}{4}$	$12\frac{1}{2}$	$84\frac{3}{8}$	$95\frac{5}{8}$	$11\frac{3}{8}$
0.55	$88\frac{1}{2}$	$102\frac{5}{8}$	$14\frac{1}{8}$	$89\frac{3}{8}$	$101\frac{7}{8}$	$12\frac{1}{2}$	90	$101\frac{3}{8}$	$11\frac{3}{8}$
0.6	$94\frac{3}{8}$	$108\frac{1}{4}$	$13\frac{7}{8}$	$95\frac{1}{8}$	$107\frac{5}{8}$	$12\frac{1}{2}$	$95\frac{3}{4}$	107	$11\frac{1}{4}$
$0.625 = \frac{5}{8}$	$97\frac{1}{4}$	$111\frac{1}{8}$	$13\frac{7}{8}$	98	$110\frac{1}{2}$	$12\frac{1}{2}$	$98\frac{5}{8}$	$109\frac{3}{4}$	$11\frac{1}{8}$
0.65	$100\frac{1}{4}$	$113\frac{7}{8}$	$13\frac{5}{8}$	$101\frac{1}{8}$	$113\frac{1}{4}$	$12\frac{1}{8}$	$101\frac{3}{4}$	$112\frac{5}{8}$	$10\frac{7}{8}$
$0.666 = \frac{2}{3}$	$102\frac{3}{8}$	$115\frac{3}{4}$	$13\frac{3}{8}$	$103\frac{1}{8}$	$115\frac{1}{4}$	$12\frac{1}{8}$	$103\frac{3}{4}$	$114\frac{5}{8}$	$10\frac{7}{8}$
0.68	104	$117\frac{3}{8}$	$13\frac{3}{8}$	$104\frac{3}{4}$	$116\frac{3}{4}$	12	$105\frac{1}{2}$	$116\frac{1}{4}$	$10\frac{3}{4}$
0.7	$106\frac{5}{8}$	$119\frac{5}{8}$	13	$107\frac{3}{8}$	119	$11\frac{5}{8}$	108	$118\frac{1}{2}$	$10\frac{1}{2}$
0.71	$107\frac{7}{8}$	$120\frac{3}{4}$	$12\frac{7}{8}$	$108\frac{5}{8}$	$120\frac{1}{4}$	$11\frac{5}{8}$	$109\frac{3}{8}$	$119\frac{3}{4}$	$10\frac{3}{8}$
0.73	$110\frac{1}{2}$	$123\frac{1}{4}$	$12\frac{3}{4}$	$111\frac{1}{4}$	$122\frac{5}{8}$	$11\frac{3}{8}$	112	$122\frac{1}{8}$	$10\frac{1}{8}$
$0.75 = \frac{3}{4}$	$113\frac{1}{4}$	$125\frac{5}{8}$	$12\frac{3}{8}$	114	$125\frac{1}{8}$	$11\frac{1}{8}$	$114\frac{5}{8}$	$124\frac{5}{8}$	10
0 76	$114\frac{5}{8}$	$126\frac{3}{4}$	$12\frac{1}{8}$	$115\frac{3}{8}$	$126\frac{3}{8}$	11	$116\frac{1}{8}$	$125\frac{7}{8}$	$9\frac{3}{4}$
0.77	$116\frac{1}{8}$	$128\frac{1}{8}$	12	$116\frac{7}{8}$	$127\frac{5}{8}$	$10\frac{3}{4}$	$117\frac{1}{2}$	$127\frac{1}{4}$	$9\frac{3}{4}$
0.78	$117\frac{5}{8}$	$129\frac{3}{8}$	$11\frac{3}{4}$	$118\frac{3}{8}$	$128\frac{7}{8}$	$10\frac{1}{2}$	119	$128\frac{1}{2}$	$9\frac{1}{2}$
0.79	$119\frac{1}{8}$	$130\frac{3}{4}$	$11\frac{5}{8}$	$119\frac{7}{8}$	$130\frac{1}{4}$	$10\frac{3}{8}$	$120\frac{1}{2}$	$129\frac{3}{4}$	$9\frac{1}{4}$
0.8	$120\frac{1}{2}$	132	$11\frac{1}{2}$	$121\frac{1}{4}$	$131\frac{1}{2}$	$10\frac{1}{4}$	$121\frac{7}{8}$	$131\frac{1}{8}$	$9\frac{1}{4}$
0.81	$122\frac{1}{8}$	$133\frac{1}{4}$	$11\frac{1}{8}$	$122\frac{3}{4}$	$132\frac{3}{4}$	10	$123\frac{1}{2}$	$132\frac{1}{2}$	9
0.82	$123\frac{5}{8}$	$134\frac{5}{8}$	11	$124\frac{1}{2}$	$134\frac{1}{4}$	$9\frac{3}{4}$	125	$133\frac{3}{4}$	$8\frac{3}{4}$
0.83	$125\frac{3}{8}$	136	$10\frac{5}{8}$	126	$135\frac{5}{8}$	$9\frac{5}{8}$	$126\frac{5}{8}$	$135\frac{1}{8}$	$8\frac{3}{4}$
0.84	127	$137\frac{1}{2}$	$10\frac{1}{2}$	$127\frac{5}{8}$	137	$9\frac{3}{8}$	$128\frac{1}{4}$	$136\frac{3}{4}$	$8\frac{1}{2}$
0.85	$128\frac{3}{8}$	$138\frac{7}{8}$	$10\frac{1}{4}$	$129\frac{3}{8}$	$138\frac{1}{2}$	$9\frac{1}{8}$	130	$138\frac{1}{4}$	$8\frac{1}{4}$
0.86	$130\frac{1}{2}$	$140\frac{3}{8}$	$9\frac{7}{8}$	$131\frac{1}{4}$	140	$8\frac{3}{4}$	$131\frac{5}{8}$	$139\frac{3}{4}$	$8\frac{1}{8}$
0.87	$132\frac{3}{8}$	142	$9\frac{5}{8}$	133	$141\frac{3}{4}$	$8\frac{3}{4}$	$133\frac{1}{2}$	$141\frac{1}{4}$	$7\frac{3}{4}$
$0.875 = \frac{7}{8}$	$133\frac{1}{4}$	$142\frac{5}{8}$	$9\frac{3}{8}$	$133\frac{3}{4}$	$142\frac{3}{8}$	$8\frac{5}{8}$	$134\frac{3}{8}$	$142\frac{1}{8}$	$7\frac{3}{4}$

TABLE

SHOWING LENGTH OF STROKE AND NUMBER OF REVOLUTIONS FOR DIFFERENT PISTON SPEEDS IN FEET PER MINUTE.

Stroke.	Speed of Piston in Feet per Minute.														
	Ft. 200	210	220	225	230	240	250	260	270	280	290	300	320	340	350
1 ft. 6 in.	67	70	73	75	76	80	83	86	90	93	97	100	106	113	116
1 " 8 "	60	63	66	68	70	72	75	78	81	84	87	90	96	100	105
1 " 10 "	55	57	60	61	63	66	68	71	74	76	79	82	88	93	96
2 " 0 "	50	52	55	56	57	60	63	65	67	70	72	75	80	85	87
2 " 3 "	44	47	49	50	51	53	55	58	60	62	64	66	72	76	78
2 " 6 "	40	42	44	45	46	48	50	52	54	56	58	60	64	68	70
2 " 9 "	36	38	40	41	42	43	45	47	49	51	53	55	58	62	64
3 " 0 "	33	35	36	37	38	40	42	43	45	47	48	50	53	56	58
3 " 3 "	31	32	33	34	35	37	38	40	41	43	44	46	50	52	54
3 " 6 "	29	30	31	32	33	34	36	37	38	40	41	43	46	48	50
3 " 9 "	27	28	29	30	31	32	33	34	36	37	39	40	43	45	47
4 " 0 "	25	26	27	28	29	30	31	32	34	35	36	38	40	42	44
4 " 3 "	23	24	25	26	27	28	29	30	32	33	34	35	38	40	41
4 " 6 "	22	23	24	25	26	27	28	29	30	31	32	33	35	38	39
4 " 9 "	21	22	23	23	24	25	26	27	28	29	30	31	33	36	37
5 " 0 "	20	21	22	22	23	24	25	26	27	28	29	30	32	34	35

PISTON-RODS.

Piston-rods, like valve-, eccentric-, and connecting-rods, are subjected to different strains, such as compression, tensile or pull, and bending; the latter is most sensibly felt in the case of engines out of line; consequently, this detail should possess sufficient strength, without extra weight of material, to resist any shock to which the engine may be subjected. Piston-rods are generally wrought-iron, though steel is frequently used, and is very much superior to the former material, not only on account of its great strength and diminished friction, but that it is less liable to become grooved by the action of the packing.

Piston-rods are in some cases connected with the cross-head by means of a screw and jam-nut; but this method of attachment is objectionable, and should never be resorted to when a key can be used; although recently some very important improvements have been made in the former mode of attachment.

CRANK-PINS.

The crank-pin being that part of the engine by which the useful effect of the steam acting against the piston is converted into work, therefore its proper proportions are of interest and importance, and should receive due consideration from the designing engineer.

Crank-pins have a tendency to become hot, because the work absorbed by the friction of the bearing is changed into heat. But this tendency to heat is independent of its diameter, it being only necessary to provide sufficient area to dissipate the heat stored up by the friction. The bearing upon a crank-pin is less than the projected area of the pin, consequently, the length of the pin that bears

should always be used in calculating the proper proportions.

Crank-pins are generally made of wrought-iron, though steel is frequently used, as it possesses many advantages over the former material, among which are strength, durability, and diminished friction; its first cost is greater than that of iron, but in point of economy it is much cheaper.

Under the influence of the connecting-rod, the piston is placed in advance of its progress, due to the crank, throughout the front stroke, and is behind its due position at all parts of the back stroke.

TABLE

(By permission, from Auchincloss' "Link and Valve Motions.")

SHOWING THE ANGULAR POSITION OF THE CRANK-PIN CORRESPONDING WITH THE VARIOUS POINTS IN THE STROKE WHICH THE PISTON MAY OCCUPY IN THE CYLINDER.

Piston Position.	Crank Angle.	Piston Position.	Crank Angle.	Piston Position.	Crank Angle.
	Deg.		Deg.		Deg.
0.1	$36\frac{7}{8}$	$0.5625=\frac{9}{16}$	$97\frac{1}{8}$	$0.813=\frac{13}{16}$	$128\frac{5}{8}$
$0.125=\frac{1}{8}$	$41\frac{3}{8}$	0.575	$98\frac{5}{8}$	0.82	$129\frac{3}{4}$
0.15	$45\frac{5}{8}$	0.6	$101\frac{1}{2}$	0.83	$131\frac{1}{4}$
0.175	$49\frac{1}{2}$	$0.625=\frac{5}{8}$	$104\frac{1}{2}$	0.84	$132\frac{7}{8}$
0.2	$53\frac{1}{8}$	0.65	$107\frac{1}{2}$	0.85	$134\frac{3}{8}$
0.225	$56\frac{5}{8}$	$0.666=\frac{2}{3}$	$109\frac{1}{2}$	0.86	$136\frac{1}{8}$
$0.25=\frac{1}{4}$	60	0.68	$111\frac{1}{8}$	0.87	$137\frac{3}{4}$
0.275	$63\frac{1}{4}$	$0.687=\frac{11}{16}$	112	$0.875=\frac{7}{8}$	$138\frac{5}{8}$
0.3	$66\frac{3}{8}$	0.69	$112\frac{3}{8}$	0.88	$139\frac{1}{2}$
0.325	$69\frac{1}{2}$	0.7	$113\frac{5}{8}$	0.89	$141\frac{1}{4}$
$0.333=\frac{1}{3}$	$70\frac{1}{2}$	0.71	$114\frac{7}{8}$	0.9	$143\frac{1}{8}$
0.35	$72\frac{1}{2}$	0.72	$116\frac{1}{8}$	0.91	$145\frac{1}{8}$
0.375	$75\frac{1}{2}$	0.73	$117\frac{3}{8}$	0.92	$147\frac{1}{8}$
0.4	$78\frac{1}{2}$	0.74	$118\frac{5}{8}$	0.93	$149\frac{3}{8}$
0.425	$81\frac{3}{8}$	$0.75=\frac{3}{4}$	120	0.94	$151\frac{5}{8}$
$0.437=\frac{7}{16}$	$82\frac{7}{8}$	0.76	$121\frac{3}{8}$	0.95	$154\frac{1}{8}$
0.45	$84\frac{1}{4}$	0.77	$122\frac{5}{8}$	0.96	$156\frac{7}{8}$
0.475	$87\frac{1}{8}$	0.78	$124\frac{1}{8}$	0.97	$160\frac{1}{8}$
$0.5=\frac{1}{2}$	90	0.79	$125\frac{1}{2}$	0.98	$163\frac{3}{4}$
0.525	$92\frac{7}{8}$	0.8	$126\frac{7}{8}$	0.99	$168\frac{1}{2}$
0.55	$95\frac{3}{4}$	0.81	$128\frac{1}{4}$	1.00	180

STEAM-CHESTS.

The steam-chest should be made as small as possible, in order to avoid radiation of heat, weakness, and large joints; but it must contain ample room for valve adjustment, and for the passage of the full volume of steam required in the cylinder from one end to the other as the valve moves. It should also be of sufficient length to admit of extra lap being added to the valve, if necessary.

VALVE-RODS.

Valve-rods, like connecting- and eccentric-rods, are subject to certain strains, among which are, most sensible compression, and tensile or pull. They are most generally made of wrought-iron; but, like piston-rods, when made of steel, are much more durable, and less liable to spring or become fluted by the action of the packing.

Length of Valve-rods.—To find the proper length of valve-rods, place the valve centrally over the ports, and the rocker or intermediate bearings in a perpendicular position; then the length of the valve-rod can be accurately determined by measuring from the centre of the rocker-stud to the centre of the valve.

GUIDES.

The guides of steam-engines embrace a great variety of designs and forms, and, as there is no rule which would apply to the guides of all classes of engines, consequently, in fixing their strength, as in many other details of the steam-engine, we must be governed entirely by practice; but they should be made of sufficient strength to prevent the possibility of springing at any speed to which the engine might be subjected. The strain on the guides

varies in differently connected engines, as with short connected engines it is very severe; while with those having long connections it is but little more than the weight of the cross-head and connecting-rod.

The strength or stiffness of guides depends more on design and shape, than on the quantity of material employed, as, by a proper distribution of the metal, they can be made sufficiently strong without being extra heavy. The form of guides in most general use are the **V** shaped, round, and flat; but the latter possesses many advantages over any other form, as they can be made more cheaply, and are less difficult to renew or repair.

Guides are sometimes cast solid with the bed-plate; this arrangement answers very well for vertical engines, but for horizontal engines, it is decidedly objectionable, and cannot be advocated on any other grounds except that of economy, as they are impossible to readjust and very difficult to renew.

ROCK-SHAFTS.

The diameter and length of the rock-shaft bearing depend somewhat on the construction of the engine; if long and subjected to torsion, or working an unbalanced slide-valve, the diameter should be from $\frac{1}{3}$ to $\frac{1}{2}$ the diameter of the engine-shaft; if not subjected to torsion, $\frac{1}{4}$ the diameter of the engine-shaft will do.

Rock-shaft Pin.—The diameter of the rock-shaft pin should not be less than the valve-stem; but if an overhanging pin, it should be from $1\frac{1}{4}$ to $1\frac{1}{2}$ the diameter of the valve-stem.

CROSS-HEADS.

The form of the cross-head, like many other details of the steam-engine, is influenced by the circumstances of

design and construction; but the most essential requisite of the cross-head is, that it should possess sufficient strength to resist the strains transmitted from the moving mass, and also have ample bearing on the guides to prevent the possibility of rapid wear or excessive friction.

STEAM-PORTS.

The dimensions of the steam-ports rank next in importance to the cut-off in their controlling influence upon the proportions of the valve-seat and face. They may justly be considered as a base, from which all the other dimensions are derived in conformity with certain mechanical laws.

Their value depends greatly upon the manner in which the ports are employed, whether simply for admitting the steam to the cylinder, or for purposes both of admission and escape.

In case of admission, if the port is properly designed, it is evident that the pressure will be sustained at substantially a constant quantity by the flow of steam from the

boiler. But with the exhaust the case is different, as the steam is forced into the atmosphere with a constantly diminishing pressure and less velocity.

When a small travel of the valve is essential, the length of the port should be made as nearly equal to the diameter of the cylinder as possible.

The ports of long-stroke slide-valve engines should be located as near the ends of the cylinder as possible, in order to reduce the cubic contents of the passages, as at each stroke of the engine the contents of long passages are thrown away, without producing any effect except to increase the volume of the exhaust. By locating the ports near the ends of the cylinder, the steam is admitted with a higher pressure than if the passages were long.

TABLE

(By permission, from Auchincloss' "Link and Valve Motions.")

SHOWING THE PROPER AREA OF STEAM-PORTS FOR DIFFERENT PISTON SPEEDS.

Speed of Piston.	Port Area.	Speed of Piston.	Port Area.
Feet per Minute.	Area of Piston.	Feet per Minute.	Area of Piston.
200	.04	450	.077
250	.047	500	.085
300	.055	550	.092
350	.062	600	.1
400	.07		

Having decided the relative area that the ports should bear to the cylinder area, it is next necessary to resolve it into its proper factors of diameter, or length and width. For instance, if the steam cylinder be 20 inches in diameter, and the port area be $\frac{1}{16}$ the area of the cylinder, if the valve be poppet or conical, the diameter of the steam-ports will require to be 5 inches; or, if proportioned for a slide-valve, for the diameter of cylinder and the same pro-

portionate area of port, it would need to be $15\frac{3}{4}$ inches long by $1\frac{1}{4}$ inches wide.

Ports should be made as straight and direct as the circumstances of design and construction will permit; it is also desirable that their surfaces should be as smooth as possible.

SLIDE-VALVES.

The slide-valve is that part of the steam-engine which causes the motion of the piston to be reciprocating. It is made to slide upon a smooth surface, called the valve-seat, in which there are three openings — two for the admission of steam to the cylinder alternately, while the use of the third is to convey away the waste steam. The first two are, therefore, termed the steam-ports, and the remaining one the eduction- or exhaust-port.

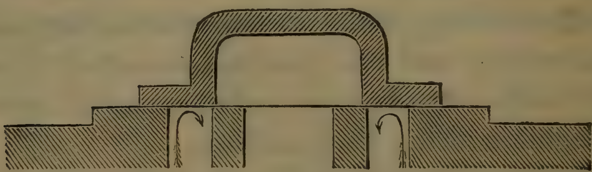
Probably no part of the steam-engine has been the subject of more thought and discussion than the slide-valve. Its proportions have been discussed in elaborate treatises, and its movements and functions analyzed by profound mathematicians, with the aid of the most extensive formulas and calculations; and yet, notwithstanding all these investigations, any one who undertakes to study its action, will find it difficult to discover anywhere a full or satisfactory explanation of the whole subject.

If some of our learned professors would instruct engineers how to design and construct a slide-valve that would give better results, under the varying circumstances to which slide-valves are subjected, than any now in use, it would do more to make them familiar with the principles involved in the construction and working of the slide-valve than any geometrical solution of its movements that might be given, however learned, as such theories are but very imperfectly understood by engineers in general.

In examining the special application of the slide-valve to the steam-engine, it will be necessary to consider what the requirements of the engine are; for the valves, of whatever kind, being to that machine what the lungs are to the body, must necessarily be so actuated as to regulate the admission and escape of the steam, which is its breath, in accordance with the conditions imposed by the motion of the piston.

The valve may be said to be the vital principle of the engine. It controls the outlet to the coal and wood pile. It is, therefore, of the highest importance that it should work practically under all circumstances.

Now the admission of steam is one thing and its escape is another, and though both may be regulated by what is called one valve, because it is made in one piece, yet this is not by any means necessary. Four separate valves may be, and sometimes are, employed in stationary engines,—a steam- and an exhaust-valve at each end of the cylinder; but the functions of all these are distinctly performed by the common three-ported slide-valve.



Position of the Slide-valve when in the Centre of its Travel.

It is evident that the admission cannot continue longer, in any case, than the stroke does, so that by the time that is completed, the valve must have opened and closed the port. These conditions determine the modification of the movement which must be used, and the greatest breadth of the port for any assumed travel of valve.

When the motion of a slide-valve is produced by

means of an eccentric, keyed to the crank-shaft and revolving with it, the relative positions of the piston and slide-valve depend upon the relative positions of the crank and eccentric.

The greatest opening of the port is half the travel of the valve; in this case the steam is admitted during the whole stroke of the piston, at the beginning of which the valve, which has no lap, is at the centre of its travel.

If the eccentric be so placed that at the beginning of the stroke of the piston the valve is not at the centre of its travel, the opening of the port will be reduced, and it will be closed before the piston completes its stroke.

In this case, the opening of the port will be less than half the travel, by as much as the valve, at the beginning of the stroke of the piston, varies from its original central position. And when the valve is at half-stroke, it will overlap the port on the opening edge to the same extent.

The point in the stroke of the piston at which the port will be closed and the steam cut off, will depend upon the angular position of the eccentric at the beginning of the stroke.

When the valve is so formed that, at half-stroke, the faces of the valve do not close the steam-ports internally, the amount by which each face comes short of the inner edge of the port is known as *inside clearance.*

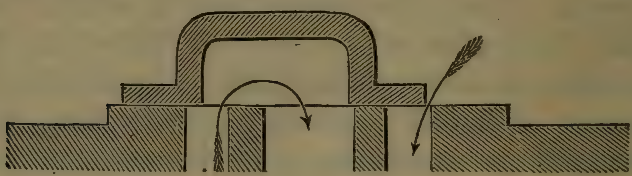
From the nature of the valve motion, it follows that the distribution is controlled by the "outer and inner edges of the extreme-ports and of the valve." The mere width of the exhaust-port or thickness of bars is immaterial to the timing of the distribution.

The extreme edges of the steam-ports and those of the valve regulate the admission and suppression; and the inner edges of the ports and the valve command the release and compression.

For every stroke of the piston, four different events occur—the admission, the suppression, the release, and the compression.

The *advance* of the *valve* denotes the amount by which the valve has travelled beyond its middle position, when the piston is at the end of the stroke, and is known as *linear advance.*

The slide-valve is more wasteful of steam than the poppet, or other forms of valve, in consequence of the long ports necessary to its use; but even with this defect, it must be conceded that nothing has as yet been introduced that has so well answered the purpose of controlling the induction and eduction of the steam to and from the cylinder as the slide-valve.

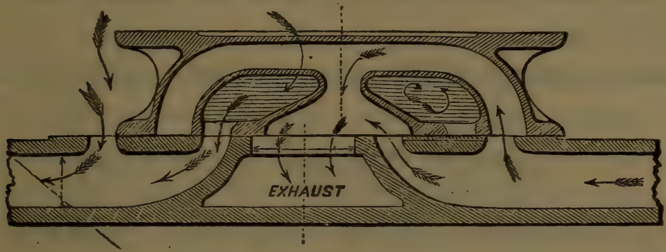


Position of the Slide-valve when the Crank is at the "Dead Centre."

When correctly designed and well-made, the slide-valve is one of the simplest and most effective devices ever invented for its office; and, on account of its simplicity of form, durability, and positive action, it has been able to successfully compete with all other forms of valves; nor is it at all likely that it will ever be superseded by any other form for engines of moderate size, more particularly where high piston speed is an object.

The friction of the slide-valve depends to a certain extent on the distance traversed by the valve. Hence it is desirable to reduce the travel as much as possible, more especially in the case of large engines. This object can be accomplished by increasing the number of ports as shown

in the accompanying cut; so that one half the travel will be sufficient to give a full port area.



Short Travel Slide-valves.

PROPORTIONS OF SLIDE-VALVES.

In order to show how to properly proportion a slide-valve, it will be necessary to give an example.

Example.—Length of valve, $8\frac{1}{2}$ inches; exhaust opening under valve, 4 inches; exhaust-port in face, $2\frac{1}{2}$ inches; inside bridge, $\frac{3}{4}$ inches thick; steam-ports, $1\frac{1}{4}$ inches wide; travel of valve, $4\frac{1}{2}$ inches; lead $\frac{1}{16}$. These proportions give a one-inch lap on each side when the valve is in the middle of its travel; the travel of this valve is 3·6 times the width of the port, which may be accepted as a good proportion for ordinary practice.

LAP ON THE SLIDE-VALVE.

The term "lap" is familiar to all steam engineers, as denoting those portions or edges of the working-faces of the valves which extend past or beyond the ports. The object of lap is to work the steam expansively; as, when the valve has lap, it cuts off the steam supply to the piston before the latter has travelled to the end of the stroke; without lap, there would be no expansion, because admission and release would occur at the same time.

Lap also induces an early and efficient release, because the lead of the exhaust, or the amount which the valve is

open to the exhaust at the end of the stroke, is increased by the amount of lap on the outside.

Lap on the steam side is termed the *outside lap*, while lap on the exhaust side is known as *inside lap.*

With a common slide-valve, it is not practicable to cut off the steam supply to the cylinder sufficiently early in the stroke to effect so large a degree of expansion as by some other means, because it would require the valve to have an excessive amount of outside lap, and the exhaust would take place too early in the stroke, thus causing the piston to travel a large proportion of the latter part of the stroke without having any pressure of steam behind it.

Slide-valves work to better advantage when the lap is so proportioned as to cut off the steam at from two-thirds to three-quarters of the stroke, than at any other point, because of the comparatively long stroke of the valve when more lap is added, and the great amount of friction generated between the valve-face and its seat.

The amount of inside lap is at all times to be governed by the speed at which the engine is to run, but it should never, in any case, be less than $\frac{1}{16}$ of an inch. Fast-running engines might have inside lap equal to one-half the outside lap, while engines travelling at slow speeds might have a little more.

The slide-valve is sometimes so proportioned as to give it inside clearance, that is, the exhaust cavity in the face of the valve is wider than the nearest edges of the steam-ports in the seat, so that, when the valve is placed centrally over the ports, there is a clear communication, to the extent of the clearance, between each steam-port and the exhaust. The object of clearance is to give the valve a freer exhaust; but this is a grave mistake, as, in proportioning a slide-valve, the inside extreme should never exceed line and line.

Inside clearance in a valve having outside lap is an evil, as it increases the exhaust opening in addition to that given by the lead of the valve, and allows the steam to escape too early in the stroke to produce the effect that it would if the valve had a certain proportion of inside lap.

POPPET OR CONICAL VALVES.

On a poppet-valve, there cannot be any lap; but the same effect is obtained in engines of this character by so adjusting the eccentrics and the lifting toes, that the valves will be lowered into their seats at the right period, or at the time when the passages require to be closed to produce the effect. The early closing of the steam-valves allows the steam to expand itself, and thereby to work with economy in the cylinder. The early closing of the exhaust-valves prevents the escape of the weak steam which is before the piston, and compresses it.

By closing the exhaust-valves very early, this compression may be carried to such an extent as to induce a pressure equal, or even superior, to that in the boiler. It is never desirable, in engines of this character, to "compress" to this extent, but by judiciously closing the exhaust-valves, and producing a gentle compression, an effect somewhat equivalent to "lead" is produced. In some engines, this compression is substituted for lead altogether, and the piston, at the moment of commencing its stroke, is only actuated by the steam which has been thus imprisoned.

This mode of operating works very economically, because it tends to save the steam, which would otherwise be required to fill the valve-passages and the little space at the end of the cylinder generally known as the "clearance." In other words, the steam, on flowing in at the commencement of the stroke, usually has to fill up a cer-

tain amount of empty space at the end of the cylinder before it begins to act with its proper effect upon the piston; but when this space has been previously filled to some extent, by the compression above described, a smaller quantity of steam is required to be drawn from the boiler for this purpose. In the Corliss engine, and some others in which the valves are opened and closed very rapidly, this mode of working is almost invariably practised.

Lift of Conical or Poppet Valves.—The required lift of conical or poppet-valves to give an opening equal to the area of the port is one-half the radius, or one-quarter of the diameter, which can be explained as follows: If a cylinder be drawn, whose height equals one-fourth its diameter, the convex surface of such a cylinder is just equal to the area of the circle of the cylinder. From this it is evident that if a circular valve of any diameter lifts from its seat a distance equal to one-fourth its diameter, the area of the opening round it will equal the area corresponding to its diameter.

Rule *for finding the Required Amount of Lap for a Slide-valve corresponding to any desired Point of Cut-off.*—From the length of stroke of piston, subtract the length of the stroke that is to be made before the steam is cut off; divide the remainder by the stroke of the piston, and extract the square root of the quotient. Multiply this root by half the throw of the valve; from the product subtract half the lead, and the remainder will give the lap required.

Another Rule.—Multiply the given stroke of the valve by the decimal numbers under each point of cut-off.

Cut-off,	$\frac{1}{2}$	$\frac{7}{12}$	$\frac{2}{3}$	$\frac{3}{4}$	$\frac{5}{6}$	$\frac{7}{8}$	$\frac{11}{12}$
Multiplier,	·354	·323	·289	·250	·204	·177	·144

TABLE

SHOWING THE AMOUNT OF "LAP" REQUIRED FOR SLIDE-VALVES OF STATIONARY ENGINES WHEN THE STEAM IS TO BE WORKED EXPANSIVELY.

The travel of the valves being ascertained, and also the amount of cut-off desired, the following table shows the amount of "lap." For instance, if a valve has $\frac{7}{8}$ lap, it will overlap each steam-port $\frac{7}{8}$ of an inch when in the centre of its travel.

Travel of the Valve in Inches.	The Travel of the Piston where the Steam is cut off.							
	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{5}{12}$	$\frac{1}{2}$	$\frac{7}{12}$	$\frac{2}{3}$	$\frac{3}{4}$	$\frac{10}{12}$
	The required "Lap."							
2	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{11}{16}$	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{3}{8}$
$2\frac{1}{2}$	$1\frac{1}{16}$	1	$\frac{7}{8}$	$\frac{13}{16}$	$\frac{11}{16}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{7}{16}$
3	$1\frac{1}{4}$	$1\frac{3}{16}$	$1\frac{1}{8}$	1	$\frac{15}{16}$	$\frac{3}{4}$	$\frac{5}{8}$	$\frac{9}{16}$
$3\frac{1}{2}$	$1\frac{1}{2}$	$1\frac{5}{16}$	$1\frac{3}{16}$	$1\frac{1}{8}$	$1\frac{1}{16}$	1	$\frac{7}{8}$	$\frac{3}{4}$
4	$1\frac{3}{4}$	$1\frac{9}{16}$	$1\frac{7}{16}$	$1\frac{5}{16}$	$1\frac{1}{4}$	$1\frac{1}{16}$	1	$\frac{13}{16}$
$4\frac{1}{2}$	2	$1\frac{13}{16}$	$1\frac{9}{16}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$\frac{7}{8}$
5	$2\frac{1}{8}$	2	$1\frac{13}{16}$	$1\frac{9}{16}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{4}$	1
$5\frac{1}{2}$	$2\frac{5}{16}$	$2\frac{3}{16}$	2	$1\frac{13}{16}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{3}{8}$	$1\frac{1}{8}$
6	$2\frac{1}{2}$	$2\frac{7}{16}$	$2\frac{3}{16}$	2	$1\frac{13}{16}$	$1\frac{5}{8}$	$1\frac{1}{2}$	$1\frac{3}{16}$
$6\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{9}{16}$	$2\frac{7}{16}$	$2\frac{3}{32}$	2	$1\frac{13}{16}$	$1\frac{5}{8}$	$1\frac{1}{4}$
7	3	$2\frac{11}{16}$	$2\frac{9}{16}$	$2\frac{3}{8}$	$2\frac{3}{32}$	2	$1\frac{3}{4}$	$1\frac{7}{8}$
$7\frac{1}{2}$	$3\frac{3}{16}$	3	$2\frac{11}{16}$	$2\frac{1}{2}$	$2\frac{3}{8}$	$2\frac{3}{32}$	$1\frac{7}{8}$	$1\frac{1}{2}$
8	$3\frac{5}{16}$	$3\frac{3}{16}$	3	$2\frac{5}{8}$	$2\frac{1}{2}$	$2\frac{3}{8}$	2	$1\frac{5}{8}$
$8\frac{1}{2}$	$3\frac{5}{8}$	$3\frac{5}{16}$	$3\frac{3}{16}$	$2\frac{13}{16}$	$2\frac{11}{16}$	$2\frac{1}{2}$	$2\frac{1}{8}$	$1\frac{3}{4}$
9	$3\frac{13}{16}$	$3\frac{5}{8}$	$3\frac{5}{16}$	3	$2\frac{13}{16}$	$2\frac{11}{16}$	$2\frac{1}{4}$	$1\frac{7}{8}$
$9\frac{1}{2}$	4	$3\frac{13}{16}$	$3\frac{5}{8}$	$3\frac{3}{16}$	3	$2\frac{13}{16}$	$2\frac{3}{8}$	2
10	$4\frac{1}{4}$	4	$3\frac{13}{16}$	$3\frac{5}{16}$	$3\frac{3}{16}$	3	$2\frac{1}{2}$	$2\frac{1}{16}$
$10\frac{1}{2}$	$4\frac{7}{16}$	$4\frac{1}{4}$	4	$3\frac{1}{2}$	$3\frac{5}{16}$	$3\frac{1}{8}$	$2\frac{5}{8}$	$2\frac{3}{16}$
11	$4\frac{9}{16}$	$4\frac{7}{16}$	$4\frac{1}{4}$	$3\frac{5}{8}$	$3\frac{1}{2}$	$3\frac{3}{16}$	$2\frac{3}{4}$	$2\frac{1}{4}$
$11\frac{1}{2}$	$4\frac{13}{16}$	$4\frac{9}{16}$	$4\frac{7}{16}$	$3\frac{7}{8}$	$3\frac{5}{8}$	$3\frac{3}{8}$	$2\frac{7}{8}$	$2\frac{3}{8}$
12	5	$4\frac{13}{16}$	$4\frac{9}{16}$	$4\frac{1}{8}$	4	$3\frac{5}{8}$	3	$2\frac{1}{2}$

LEAD OF THE SLIDE-VALVE.

The object of the lead is to enable the steam to act as a cushion against the piston before it arrives at the end of the stroke, and cause it to reverse its motion easily; and

also to supply the steam of full pressure to the piston from the instant it has passed its dead centre.

When the work an engine has to perform is very irregular, as is generally the case in rolling-mills, or if the different parts of the engine be badly worn or have much lost motion, considerable lead is absolutely necessary, in order that the steam admitted will offer an opposing and gradual force in a direction opposite to that in which the engine is moving, and take up the play in the different parts before the piston has reversed its motion.

If the piston, after passing the centre, should meet with no opposing force, it would travel very fast during the time in which the play was being taken up, and, when the valve opened again, it would receive a check from the action of the steam which would cause it to thump or pound.

Lead, like many other details, requires the exercise of mechanical skill and judgment, as, if a valve has too much lead, not only is there a great loss of power, but the piston receives a violent shock at each end of the stroke, and it will be found almost impossible to keep the packing tight around the piston-rod in consequence of the excessive cushioning.

If the amount of lead be so great as to admit steam of the full pressure to the passages and clearance, the piston will have to force it back into the passages and chest, exposing the wrist- and crank-pins to a fearful shearing strain when the crank is at its weakest point — the fly-wheel travelling fast and the piston moving very slowly.

No general rule can be given for the amount of lead that would be best suited or most advantageous for all classes of engines, as that must be determined by the circumstances of construction, speed, work, etc.

In the case of vertical engines having the cylinder

above and the crank below, it is customary to give less lead on the upper than on the lower port, as the wear in the valve connections has a tendency to increase the lead on the upper end. With vertical engines having the crank above and the cylinder below, these conditions are reversed. It is also customary in the case of horizontal engines to give more lead on the front than on the back end, in consequence of the reduced capacity of that end arising from the space occupied by the piston-rod.

For stationary engines, the lead varies in general practice from $\frac{1}{32}$ to $\frac{3}{16}$ of an inch; the exhaust lead being in all cases double the amount of steam lead.

The average amount of lead, in full gear, for freight locomotives, is $\frac{1}{12}$ of an inch; for locomotives running accommodation passenger trains, $\frac{1}{11}$ of an inch; and fast express locomotives, $\frac{1}{10}$ of an inch.

CLEARANCE.

The term clearance is used to express the extent of the space which exists between the piston, the cylinder-head, and the valve-face at each end of the stroke. For each stroke of the piston, this space must be filled with steam, which in no way improves the action of the engine, but rather increases the amount of steam to be exhausted on the return-stroke.

It is, therefore, an object of great importance, in point of economy, to have the valve-face as near the bore of the cylinder as possible, in order that the cubic contents of the space in the cylinder unoccupied by the piston and the steam passages may be reduced to their smallest capacity.

COMPRESSION.

Compression is the term used to express the distance the piston moves in the cylinder after release or exhaust has taken place, and the exhaust passage closed by the return-stroke of the valve, whereby the communication is cut off from the exhaust-port and that end of cylinder. Compression takes place between the piston and cylinder-head at each end of the stroke, and the distance from the end of the cylinder at which it takes place depends on the amount of lap on the valve.

FRICTION OF SLIDE-VALVES.

All engineers agree that there is a great loss of power in working the slide-valve, but differ in the amount, from the fact that no correct data has been formed by which to make such calculations; but an idea has been very generally entertained by engineers, that the number of square inches in a slide-valve, and the pressure of steam in pounds per square inch, represented the total pressure on its back, or that the pressure was equal to the pressure of steam per square inch on the back of a valve minus the area of the steam-ports.

Such conclusions, however, are erroneous, as the number of square inches in a slide-valve, and the pounds pressure per square inch, would only represent the weight on its back, if we consider the valve as a solid block of iron with a smooth surface, resting on a smooth solid bearing perfectly steam-tight, as then the steam would press on every square inch of surface with the same force that a dead weight laid upon it would.

These conditions are never found in a slide-valve except in one position — that one, when the valve overlaps

both ports, and the engine is at rest. As soon, however, as the valve is moved, the steam enters the open port, and the pressure is practically taken off that end of it.

When the valve is moved back over the port, the steam that is shut up within the cylinder will press up against the under side of the valve-face with a force exactly equal to the pressure at the point in the stroke of the piston at which the valve closed.

As the valve continues its stroke, the other port will be opened, and the steam that was shut up in the cylinder begins to exhaust; and the pressure against the under side of the valve will be the same as the pressure in the cylinder at the end of the stroke. This pressure is only for a brief period, for in engines with well-proportioned steam-ports, the time occupied in exhausting the contents of the cylinder is very short.

While the steam is entering the open port, and after the exhaust has passed through the closed port, the pressure on the under side of the valve will be just the ordinary back pressure.

Therefore, in order to determine the pressure on a slide-valve, we must consider the pressure in the cylinder at the time of cutting off the back pressure against the piston, the area of the ports, etc.

Rule for finding the Pressure on Slide-valves.—Multiply the unbalanced area of the valve in inches by the pressure of steam in pounds per square inch; add the weight of the valve in pounds, and multiply the sum by 0·15.

Another Rule.—Multiply the combined area of the bearing surface and ports in inches by the steam pressure in pounds per square inch on the back of the valve; multiply this product by the coefficient of friction between the

two surfaces.* The product will be the force required to move the valve when unbalanced.

BALANCED SLIDE-VALVES.

The mechanical difficulties heretofore experienced in producing a balanced slide-valve that would be reliable under all the varying circumstances to which slide-valves are exposed, have not, so far, been fully overcome, as may be inferred from the views of the American Railway Master Mechanics, expressed at their last annual convention at Chicago. Their discussions on this subject were of great importance, and embraced a wider range of experience than any other equal number of mechanics in this country. While they all agreed that the use of balanced slide-valves on locomotives would be of great benefit, as they would materially diminish the wear of valve gear; they were almost unanimously of the opinion that not one of the balance valves now in use on locomotives was perfectly reliable, as they were all open to the objection of leaking, and the expense incurred by the use of the unbalanced valve was less than that of keeping the balance valve in repair.

The removal of the weight from the back of the valve would be a step in the right direction, and, as the attention of engineers and railway mechanics is directed that way, and the difficulty does not appear to be insurmountable, it is not at all improbable that some of the different forms of balance valves now in use will be so modified and improved as to accomplish the desired object.

There are many forms of the balance valve that have rendered good service, but none of them have, so far, met all the requirements of a good steam-tight slide-valve.

* See Table of Coefficients, pages 480, 481.

FITTING SLIDE-VALVES.

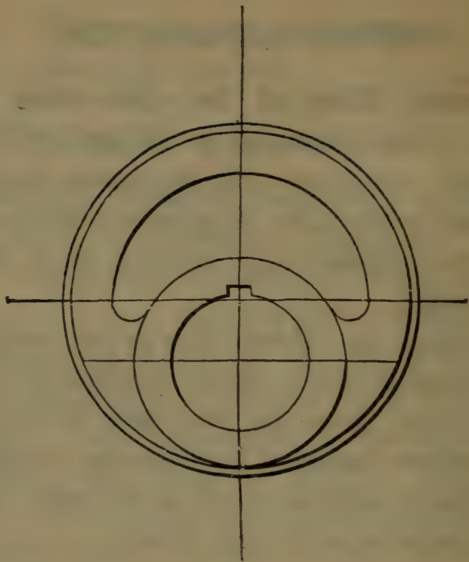
The accurate fitting of the slide-valve to its seat requires the closest attention on the part of the engineer and machinist; yet these very important details are very often neglected under the impression that the valve will wear down to a steam-tight bearing, or, in other words, find its own seat.

But experience has shown this to be a very erroneous idea, as improperly fitted valves frequently commence to wear out of shape from the first time they are used, and the extra expense incurred in replacing them frequently amounts to three times as much as it would have cost to make a thorough fit in the first place.

SLIDE-VALVE CONNECTIONS.

The most ordinary methods of connecting the slide-valve are by means of an oblong hole in the back of the valve, through which the rod is slipped and secured at each end by jam-nuts; or by a single nut resting in a recess formed in the back of the valve; or, as in the case of the locomotive, by means of a yoke which encircles the valve; the latter mode forms the most permanent attachment, particularly for large engines, as the jam-nuts, though affording the best facilities for adjustment, are objectionable on account of their liability to become loose and wear the stem; the single nut, unless thoroughly fitted, is open to the same objection.

It is no uncommon thing to find well-proportioned and well-fitted slide-valves, only a short time in use, worn rounding on the face in consequence of the vibration or springing of defective connections.



ECCENTRICS.

The term "eccentric" is applied to all such curves as are composed of points situated at unequal distances from a central point or axis.

The motion imparted to the slide-valve is generally derived from two principles of action — vibratory and rotary. When the former is the prime mover, the speed of the valve is the same throughout the stroke; or rather, if the motion is imparted by the piston, the motion of it and the valve would be equal.

Rotary motion is more often adopted than any other for the transmission of power and action, and at present, small cranks and eccentrics are the prevailing means employed to impart the motion required for the slide-valve. The speed of the crank and the eccentric are proportionately the same in theory and practice.

Upon inspection, it will be seen that the eccentric is only a mechanical subterfuge for a small crank. This being so, a crank of the ordinary form may, and frequent-

ly is, used instead of an eccentric — the latter being a mechanical equivalent introduced, because the use of the crank is, for special reasons, inconvenient or impracticable.

Since the shaft to which the eccentric is fixed makes a half revolution while the piston is making one stroke, it follows that whatever device may be used for converting the reciprocating motion of the piston into rotary motion, the slide-valve may be actuated by an eccentric fixed on any shaft which makes a half revolution at each stroke of the piston.

It will now be observed that the eccentric and valve connection is nothing more nor less than a small crank with a long connecting rod; the valve will therefore move in precisely the same manner as the piston, and will have, in its progress from one extremity of the travel to the opposite, like irregularities.

In other words, when the eccentric arrives at the positions for cut-off and lead, the valve will be drawn beyond its true position—measured towards the eccentric—by a distance dependent on the ratio between the throw of the eccentric and the length of its rod.

When the eccentric stands at right angles to the crank, the exhaust closes and release commences at the *extremities* of the stroke; consequently, if the eccentric be moved ahead 30°, not only will the cut-off take place 30° earlier, or at a crank-angle of 120° instead of 150°, but the release will take place 30° earlier, or at the 150° crank-angle.

For a cut-off, say of 140°, there would be required an angular advance of 20°, and a lap equivalent to the distance these degrees remove the eccentric centre from the line at right angles to the crank; for a cut-off of 160°, an advance of 10°, with a corresponding lap, and so on, the exhaust closure taking place respectively at the 160° and 170° crank-angles.

This closure of the exhaust confines the steam in the cylinder until the port is again opened for the return stroke; consequently, the piston in its progress will meet with increasing resistance from the steam, which it thus compresses into a less and less volume.

Such opposition, when nicely proportioned, aids in overcoming the momentum stored up in the reciprocating parts of the engine, and tends to bring them to a uniform state of rest at the end of each stroke.

Since the closure of one port is simultaneous with the opening of the other, a release will take the place of the steam which was previously impelling the piston.

Within certain limits an early release is productive of a perfect action of the parts, for an early release enables a greater portion of the steam to escape before the return stroke commences; whereas, a release at the end of the stroke would be attended by a resistance of the piston's progress, from the simple fact that steam *cannot* escape instantaneously through a small passage, but requires a certain definite portion of time, dependent on the area of the opening and the pressure of the steam.

Angular Advance.—By angular advance is meant the angle at which the eccentric stands in advance of that position which would bring the slide-valve amid stroke, when the crank is at the dead centres.

Throw or Stroke of the Eccentric.—The throw of the eccentric is twice the width of one steam-port, with twice the amount of lap on one side of the valve added.

How to find the Throw of any Eccentric.—Measure the eccentric from the shaft, on the heavy and light sides; the difference between the two is the throw.

Eccentrics of Marine Engines.—Eccentrics of marine engines are generally made in two pieces and bolted together, and, when only a single eccentric is used, are

always loose on the shaft. The eccentric is fitted on the shaft so that it can move half-way round; there are two stops on the eccentric and one on the shaft. The shaft revolves without the eccentric until it has moved a half revolution, when the pin on the shaft comes in contact with one of the stops on the eccentric, and moves it forward in the direction of the motion.

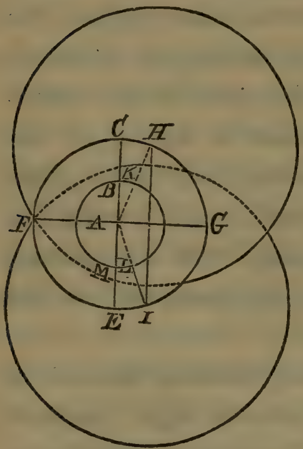
When it becomes necessary to reverse the engine, the engineer notices whether the piston is moving up or down; if moving up, he takes the starting-bar, throws the eccentric-hook out of gear, and admits steam to the top of the piston, which immediately changes the motion of the engine; and when the shaft has moved around a half revolution, the stop on the shaft comes in contact with the second stop on the eccentric, and reverses its position on the shaft.

Formula *by which to find Positions of Double Eccentrics on the Shaft.*—Draw upon a board two straight lines at right angle to each other, and from their point of intersection as a centre describe two circles, one representing the circle of the eccentric, the other the crank-shaft; draw a straight line parallel to one of the diameters, and distant from it the amount of lap and lead; the points in which this parallel intersects the circle of the eccentric are the positions of the forward and backing eccentrics.

Through these points draw straight lines from the centre of the circle, and mark the intersection of these lines with the circle of the crank-shaft; measure with a pair of compasses the chord of the arc intercepted between either of these points and the diameter which is at right angles with the crank, the diameters being first marked on the shaft itself; then by transferring with the compasses the distances found in the diagram, and marking the points, the eccentrics may at any time be adjusted without difficulty.

Example. — Let F G and E C be the two straight lines at right angles to each other; the circle described with A B as a radius be the end view of the shaft; the circle described with A C as a radius be the circle described by the centre of the eccentrics; and H I the line parallel to E C, and distant from it the amount of the lap and lead.

Then, if F G represents the direction of the crank when on the centre, H and I will be the positions of the centres of the eccentrics, according to the rule. If, then, the points K and L, in which the lines A H and A I intersect the circle representing the shaft, be transferred to the shaft, by laying off on its end the two diameters, and the chords B K and L M, the eccentrics can readily be set.



Whether the engine be vertical, horizontal, or inclined, the eccentric occupies the same position on the shaft. The wide part, or throw, and the crank are always at right angles to each other, excepting the departure from a right angle which the lap and lead make necessary, as shown by the lines C and E and H and I. To account for this position, it must be understood that the eccentric must commence to open the port before the crank reaches the centre, or, in other words, the eccentric must commence its stroke a little ahead of the crank.

ECCENTRIC-RODS.

Eccentric-rods, like all other vibrating rods subjected to different strains, direct and angular, compression and tensile, or pull, are frequently of great length even in

engines of moderate size; consequently, rigidity is required, in order that a steady motion may be transmitted from the eccentric to the valve. A long eccentric rod is desirable.

Length of Eccentric-rod.—The length of the eccentric-rod is the distance between the centre of the crank-shaft and the centre of the rocker-pin, when the latter stands plumb.

How to Adjust the Eccentric-rod.—Place the crank on the dead centre and the eccentric at 90°, or at right angles with the crank; then adjust the eccentric-straps and place the rocker in a perpendicular position. If the eccentric-catch adjusts itself to the rocker-pin without moving the latter in either direction, the eccentric is of the proper length.

Modes of attaching Eccentric-rods.—In some instances, the end of the eccentric-rod is turned tapering, to fit a corresponding hole in a sleeve cast on the forward strap of the eccentric, in which it is secured by a vertical key; while in others, the rod is slipped loose into the sleeve and secured at both ends by jam-nuts: this arrangement affords the best facilities for adjusting the rod to its proper length.

CRANKS.

Cranks may be divided into two classes, single and double,—the former being most generally used for stationary engines, while the latter are more applicable to the engines of river-boats and sea-going steamers. Cranks are subjected to different strains, among which are most prominent, bending or deflecting, compression, and tensile. They should be considered as beams supported at one end and the load at the other; consequently, the sectional area of the crank depends on the form and the length be-

tween the centre of the crank-pin and the centre of the shaft.

All single rotative engines are provided with heavy parts, such as fly-wheel, disk-cranks, and counter-weights, which also have a rotary motion when the piston is in action. These heavy parts acquire energy during the stroke to continue the motion past the centres, where the pressure on the piston produces no effective pressure on the crank-pin.

The crank being the means most generally used for the convertion of reciprocating into rotary motion, the question has frequently arisen, whether or not there is a loss of power connected with its use; but upon examination, it will be found that the only loss of power incurred by the use of the crank is that absorbed by friction, which is the coefficient of loss in the transfer of all forces. The idea of a loss of power in the crank arose from the common error of confounding power and pressure, and forgetting that a small force exerted over a great distance in a given time may develop as much power as a large force exerted over a small distance in the same time.

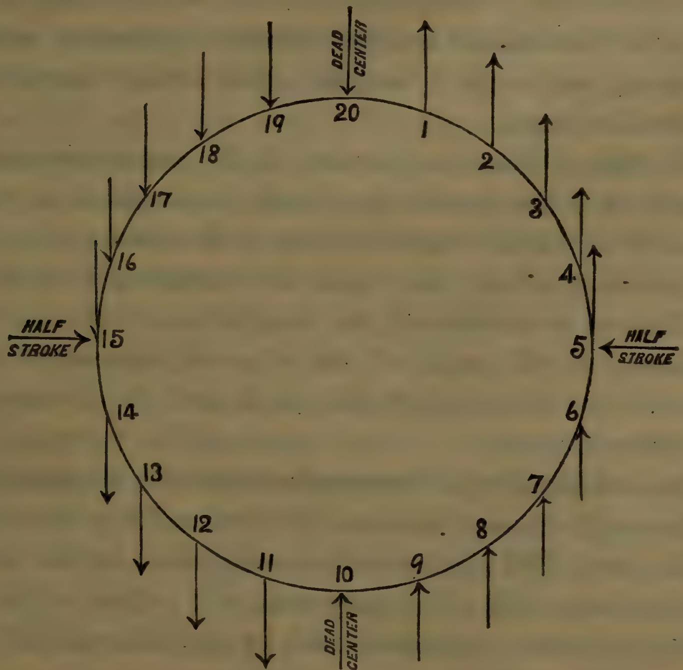
An examination of the connecting-rod of an engine in motion, will show that the two ends pass over different spaces in a given time. If, for instance, in one stroke, the end of the connecting-rod that is attached to the cross-head moves through one foot, the end which is attached to the crank-pin, and makes a half revolution in the same time, passes through 1·5708 feet.

Suppose that an engine is placed with its crank on the centre, and steam is admitted; no motion will be produced, and, consequently, there will be no power developed, and no expenditure of steam. But let the piston make a stroke; the power exerted is equal to the force or pressure acting on the piston multiplied by the space passed

through, or it will be 100 foot-pounds, assuming the data of the preceding instance.

During the same time, the crank-pin has passed through a space of 1·5708 feet, and the force or pressure exerted has been 63·66 pounds, so that the power exerted during this time, or the product of 1·5708 multiplied by 63·66 pounds, is 100 foot-pounds. Consequently, there is no loss of power in the use of the crank, all the power exerted on the piston being imparted to the crank.

Examination of the Principles Involved in the Use of the Crank.—With a pair of compasses describe a circle; draw a line through the centre, from one point of the in-



tersection of this line with the circle; divide the latter into 20 equal parts, 20 and 10 occurring at these points. Now suppose that the constant pressure of the steam in the cylinder be represented by 100, this pressure is communicated

to the crank by means of a connecting-rod, and we will suppose that the above circle is the circle described by the crank-pin, 10 and 20 coinciding, and with the division 10 forming a right line with the centre of the crank-shaft and the centre of the cylinder; of course, the points 20 and 10 are the two points where the pressure in the cylinder has no effect in turning the crank, called the "dead-points."

The points 5 and 15 are the points at which the effect of the pressure on the piston is a maximum, decreasing each way to zero. Supposing, for simplicity, the direction of the connecting-rod to remain parallel with its first position, then the effect of any pressure communicated by it to the crank-pin is resolvable into two — one acting on the centre of the crank, and, of course, inoperative towards producing motion in it, and the other acting tangentially to turn the crank.

The first of these is greatest at the commencement, or 20 and 10 of the circle of the crank, and least at the points 5 and 15; while the second is least at the first-named points and greatest at the last; and this variation is (from the well-known principles of the composition and resolution of forces) in the ratio of the sines of the angle made between the direction of the crank and the direction of the connecting-rod.

The subdivision of the crank-circle into 20 parts gives as the angle of each division 18°, and calling the radius of this circle 100, the sines of the respective angles formed by the crank and connecting-rods will represent the percentage of power communicated by the latter to turn the former. Thus:

Crank-pin at	0	0°		0.0
" " "	1	18°	Sin.	30.90
" " "	2	36°	"	58.78
" " "	3	54°	"	80.90
" " "	4	72°	"	95.11
" " "	5	90°	"	100.
" " "	6	108°	"	95.11
" " "	7	126°	"	80.90
" " "	8	144°	"	58.78
" " "	9	162°	"	30.90
" " "	10	180°	"	0.0
			Mean power,	63.11

The pressure of the steam on the piston forces the connecting-rod twice the length of the diameter of the circle in the same time that the crank-pin travels through a space equal to the whole circumference of this circle; and as the circumference of this circle bears to twice its diameter the ratio of 100 to 63·6, it follows that the pressures on the crank and piston are inversely as the space through which they move. The effects of moving powers may be represented, for comparison, by the product of the pressures into the spaces described in the same time.

The power of the steam in the cylinder being 100, and moving through a space represented by 2, we may represent it by 200; and the mean pressure on the crank, as shown above, being 63·11, moving through a space represented by 3·1416, we may represent its effect by their product, 198·26, differing but 1·74 from the power given out by the steam in the cylinder. This difference will appear smaller and smaller, according as we multiply the number of points in the circle, from which we calculate the mean pressure on the crank.

The two foregoing formulas, introduced for the purpose

of showing that the crank is no consumer of power beyond the friction incidental to the motion of all machinery, but gives out all the power it receives from the steam, are somewhat different in their conclusions, being made by different calculi; but they are both approximately correct.

CRANK-SHAFTS.

The crank-shaft of an engine is the transmitter of the power developed and expended at its middle or extremities, as the case may be. As the force exerted by the steam in the cylinder is thrown on the crank-pin, from thence to the crank, and in turn to the shaft, evidently, the strains imposed are not alike, — the pin is subjected to a shearing strain; the crank, to a deflecting or bending strain; and the shaft, to torsion and shearing strains.

Crank-shafts may be made of either cast- or wrought-iron, and, when well proportioned, the former material answers very well, especially for large engines; but when lightness and strength become an object, wrought-iron is the material generally used.

PILLOW-BLOCKS, OR MAIN BEARINGS.

The pillow-blocks, or main bearings, of a steam-engine rank next in importance to the cylinder and piston; and as they are subjected to very unequal wear and strains, unless well proportioned and thoroughly fitted, they are liable to heat, and become a continued source of annoyance. Of course, by reducing the friction, the tendency to heat is also reduced. This can be done by making the revolving surfaces smooth and using a good lubricant; but at very high pressures the lubricant is forced out from between the surfaces; the pressure at which this occurs is supposed to be about 2000 pounds to the square inch of bearing surface.

FLY-WHEELS.

The applications of fly-wheels are very general, as there are few stationary engines without them; and it has also become quite common to communicate the power directly from the periphery of the fly-wheel, by using it as a pulley or drum, and transmitting the power through a belt or band.

The fly-wheel is used as a regulator, or equalizer of motion, wherever either the power communicated or the resistance to be overcome is variable.

In the one case, the fly-wheel may be said to be a distributer of power. The communicated impulses act on the mass in motion, and go to preserve the momenta, without disturbing very sensibly the regularity of movement. The effect of one impulse is so absorbed or distributed in the momentum of the wheel, that its effect may be said to have hardly been diminished before the next impulse is received.

In the other case, or where the fly-wheel is used to overcome a variable resistance, it may be considered a collector of power; the power having been employed to get up the speed in the fly-wheel only, is retained in the mass in movement; and the whole of the power expended, with the exception of that which has been lost through friction and resistance of the air, can be brought to bear at any instant upon the resistance to be overcome.

When the crank and connecting-rod are in one straight line, as they must be twice in each revolution, the crank is said to be on its dead centre, because there the force of the piston is dead or ineffective. It is evident, that when the crank is at right angles to the connecting-rod, the latter has the most power on the former; but when the forward or backward dead centre is reached, there is no reason why

it should not remain there; but the action of the fly-wheel then shows itself, for, having on it a certain accumulated velocity, it cannot stop, but goes forward, carrying with it the crank over the dead centres.

Thus, through the momentum of the fly-wheel, no perceptible variation occurs in the velocity of the engine; the unequal leverage of the connecting-rod is also corrected, and a steady and uniform motion produced.

The fly-wheel, as before stated, is a regulator and reservoir, and not a creator of motion. The accumulated velocity in the fly-wheel, where the motion is required to be excessively equable, should be about six times that of the engine when the crank is horizontal.

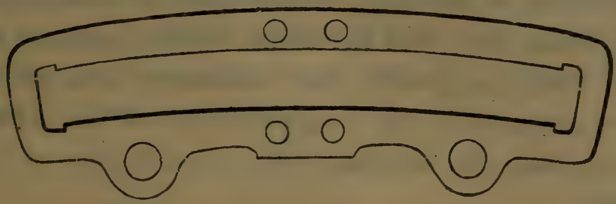
As regularity of motion is of much greater importance in some cases than in others, the weight and diameter of the fly-wheel must depend on the work and the character of the machinery the engine is intended to drive; so that in proportioning a fly-wheel to a given engine, attention must be paid to many particular circumstances rather than to any given rule.

There are circumstances in which the use of a fly-wheel may be dispensed with: where a pair of engines work side by side, whose cranks are at different angles, so that one assists the other to pass the centres, or where smoothness of motion is not an absolute necessity.

LINK-MOTION.

Were it only requisite to provide a means of starting and stopping an engine, the mechanical appliance might be very simple, but when reversing must be accomplished by the same gear, a very different mechanical arrangement becomes necessary. By means of the link the engineer is able at will to change the direction of the

engine, with only the loss of time required for overcoming the momentum of the moving parts and developing the like in a reverse direction.



A link operated by two fixed eccentrics forms, when properly suspended, an exact mechanical equivalent of the movable eccentric. Unlike the latter, however, its motion is capable of an accurate adjustment, which practically obviates the effect of irregularities in cut-off and exhaust closure, attributable to the angularity of the main connecting-rod.

Horizontal motion, communicated to the link by the joint action of the eccentrics, is a minimum at the centre of its length, where it is equal to twice the linear advance, and it increases towards the extremities of the various periods of the block in the link, or of the link on the block, on the general principle that admission varies with the travel of the valve.

The nature of the motion derived from the link is modified by the positions of the working centres, and most especially of the centres of suspension and connection; the centre of suspension is the most influential of all in regulating the admission, and its transition horizontally is much more efficacious than a vertical change of place to the same extent.

As the vertical movement of the body of the link with the consequent slip between the link and the block is the least possible when the suspended centre lies in the centre line of the link, increasing as the centre is moved later-

ally, the centre line of the link is, in this respect, the most favorable locality for the suspension, though not always practicable for equal admissions.

In practice it has been found that the stationary and shifting links have not the same neutral centres of suspension; that, in general, the stationary link should be hung by a centre in the neighborhood of the middle of its length, and the shifting link towards one of the extremities.

The utmost period of expansion obtained by a stationary link in mid-gear is 38 per cent. for 12 per cent. of admission, in which case the steam is cut off at less than one-eighth of the stroke, and expanded into a volume of 50 per cent., or one-half stroke,—4 times the initial volume, exclusive of clearance,—after which it exhausts during the remaining half-stroke.

With the stationary link the shortest admission is 11 per cent., or one-ninth of the stroke, expanding into 50 per cent., or 4½ times the initial volume, before the release takes place.

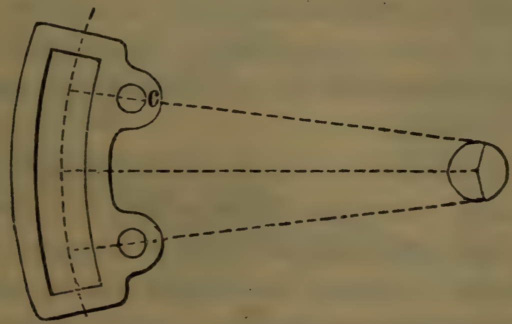
With the shifting link, the smallest attainable admission is about 17 per cent., or one-sixth of the stroke; this is about one-half more than what is obtained by the stationary link, the difference being due to the excess of lead yielded by the shifting.

As the release takes place at half-stroke, the shifting link cannot expand the steam above three times its initial volume, exclusive of clearance.

The average period of admission in full gear does not exceed 75 per cent., or three-fourths of the stroke. The admission may, however, be increased by forcing the mechanism of the valve beyond full gear — that is, by causing the block to work in the extreme overhung parts of the link, which must be extended for the purpose

beyond the centres of connection; by this expedient the throw of the valve is increased.

The periods of expansion and release increase as those of admission are diminished. The motion of each eccentric prevails in that half of the link to which it is coupled, and at the centre the motion of the link is equally composed of the two eccentrics.



The radius of the link is the length from C to the centre of the eccentric, or the horizontal distance from the centre of the shaft to the centre of the link.

Locomotives and screw-engines have on their main shaft two eccentrics and eccentric-rods for working the slide-valve. One of these eccentrics is for the forward or progressive, the other for the backward or retrogressive motion of the engine, and the same act which brings one into operation throws the other out.

When the link is moved so as to bring the valve-stem pin into its centre position, that pin then becomes a pivot, without any rectilinear, reciprocating, or other motion whatever, being entirely at rest. When the valve-stem pin is thus in the centre of the link, the valve has no motion, but is at rest in the centre of the steam-chest covering all the three passages, which shuts out the admission of steam to the cylinder.

It follows that so long as the pin holds this central

position in the link, the engine cannot move by steam; for, however it may be in the chest, it cannot reach the cylinder to act on the piston. Should it become necessary, for the purpose of adjusting any part of the machinery, to revolve the shaft by hand, it can be done with perfect safety, providing the link is maintained in a central position.

The nearer the valve-stem pin is brought to either end of the link or either eccentric-pin, the greater will be the travel of the valve, and the more the steam- and exhaust-ports will be opened.

When the forward eccentric-rod is brought nearest to the valve-stem, the engine under steam will move ahead; when the backward eccentric-rod is brought nearest the valve-stem, the motion of the engine will be reversed.

When an engine is standing under steam, the link should, in all cases, be placed at mid-gear, or, in other words, in a central position, so that the backward and forward eccentric-pins will be at equal distances from the valve-stem,—a necessary precaution to guard against accident.

To start ahead, move the link until the forward eccentric-pin comes nearest to the valve-stem.

To back, bring the backward eccentric-pin nearest to the valve-stem.

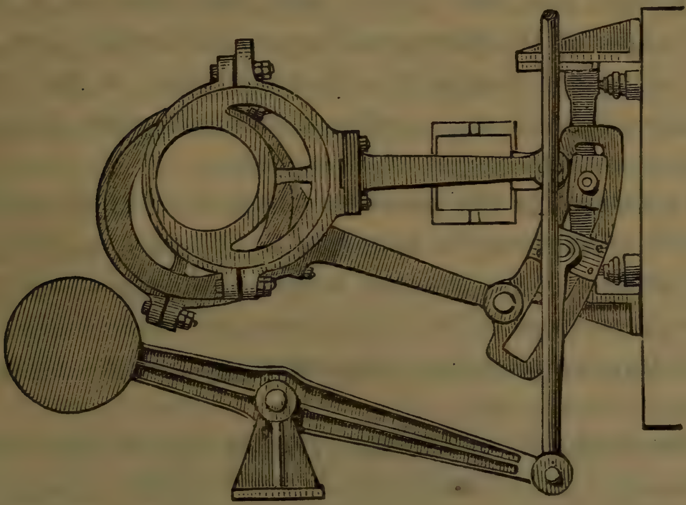
To stop, bring the two eccentric-pins to equal distances from the valve-stem.

To go ahead fast, move the link to full gear, or until the link-block is in the extremity of the link.

To go slow, bring the centre of the link near the centre of the link-block.

The following cut represents the form of link generally used on marine engines: the link is raised and lowered by means of a screw, which receives its motion from a

pair of mitre-gears on the hand-wheel shaft; the weight of the link, reach-rod, and sliding-block is counterbal-



anced by the lever and balance-weight below the motion; and thus it will be seen that the power required to shift the slide-valve is greatly reduced.

PROPORTIONS OF STEAM-ENGINES ACCORDING TO THE BEST MODERN PRACTICE.

Before any correct formula, by which to determine the proper proportions for steam-engines, can be deduced, there are many things to be considered; permanent load, weight of moving material, nature of motion, etc.

The load on the piston-rod consists of the piston at one end and the cross-head at the other; consequently, the greater the length between these two points, the more the rod is affected. For this reason, it is obvious that, when it becomes necessary to fix the area of the piston-rod, the pressure, area of cylinder, load, and length of travel must be duly considered.

The connecting-rod being hung between a sliding and a

rotary motion, the load is in some measure due to the length of the rod in proportion to the circle described. In the first case, the sliding point has a load on it due to the weight of the piston-rod beyond the stuffing-box, with the additional weight of the cross-head; in the second instance, the rotating surface is affected by the weight of the rod and the weight of the crank.

To determine the diameter of the crank-shaft, we must take into account the length of the crank as a lever, and the pressure of steam as the weight on the end of the same. The proportions of the crank-pin are likewise modified according to pressure, permanent load, length of stroke, shearing strain, etc.

The thickness of a steam cylinder may be found by the following rule.—Divide the diameter of the cylinder plus 2 by 16, and deduct a $\frac{1}{100}$ part of the diameter from the quotient, the remainder will be the proper thickness.

The depth of the piston-rings should equal $\frac{1}{4}$ the diameter of the cylinder.

The thickness of the follower-plate should be the same as that of the cylinder.

The whole thickness of the piston will therefore be $\frac{1}{4}$ the diameter of the cylinder plus twice its thickness obtained by the rule above.

The diameter of the piston-rod should be from $\frac{1}{5}$ to $\frac{1}{6}$ that of the cylinder for high-pressure engines, and $\frac{1}{7}$ for condensing engines.

The diameter of the crank-shaft may be about $\frac{4}{10}$ that of the cylinder if of wrought-iron, or $\frac{5}{10}$ if of cast-iron; but it should be $\frac{5}{10}$ of wrought-iron if extra strength be required.

The length of the crank-shaft bearing should be equal to $1\frac{1}{2}$ times its diameter, and in some cases it should be twice.

The diameter of the crank-pin should be from ·2 to ·25 that of the cylinder. Its length should be from ·275 to ·35 the diameter of the cylinder.

The diameter of the wrist- or cross-head pin should be equal to that of the crank-pin, and its length the same.

The diameter of the connecting-rod, in the neck, should equal that of the piston-rod, and should increase $\frac{1}{4}$ inch in diameter, to the foot, from the neck to the middle.

The diameter of the eccentric-rod in the neck should be $1\frac{1}{4}$ times the diameter of the valve-rod, and should increase $\frac{1}{8}$ inch in diameter to the foot of the eccentric.

The diameter of the valve-rod should be from $\frac{1}{12}$ to $\frac{1}{10}$ that of the cylinder.

The diameter of the boss of the crank, if cast-iron, should equal twice that of the shaft-journal. Its depth should equal the diameter of the shaft-journal multiplied by 7.

The diameter of the crank at the pin should equal twice the diameter of the pin, and its depth at the pin should equal the diameter of the pin multiplied by 12.

The thickness of the web of the crank should equal three times the diameter of the shaft-journal.

The boss of the crank, if of wrought-iron, should equal the diameter of the shaft-journal or the pin multiplied by 4.

The thickness of the crank should equal the diameter of the shaft-journal multiplied by 6.

The area of the crank at the centre should equal that of the shaft.

The thickness of the straps should be equal to ·44 of the diameter of the pins; but for engines requiring great strength, they ought to be $\frac{1}{2}$ the diameter of the pins.

The breadth of the strap should equal 1·1 times the diameter of the pin plus $\frac{1}{16}$.

The distance from the slot to the end of the strap should equal ·06 of the diameter of the pin.

The breadth of the gib and key should equal 1·1 times the diameter of the pin; the thickness should equal ·3 that diameter; the clearance should equal $\frac{1}{2}$ the diameter of the pin plus 2 divided by 16; the distance from the key-slot to the end of the block should equal ·44, the diameter of the pin.

The diameter of the steam-pipe should be the same as that of the crank-pin, or from ·2 to ·25 the diameter of the cylinder.

The diameter of the exhaust-pipe should be from ·25 to ·3 the diameter of the cylinder.

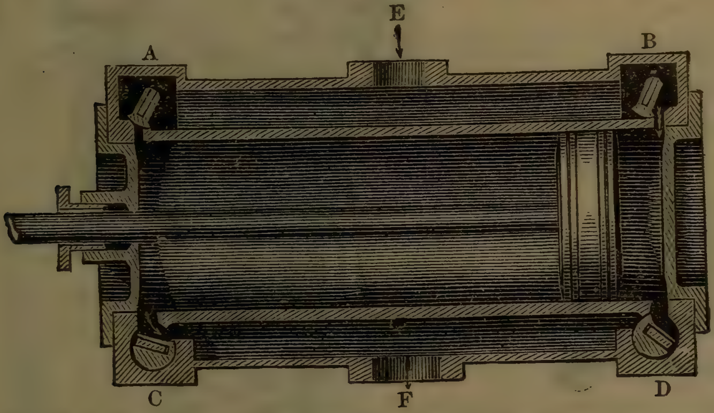
The length of the cross-head bearings should be equal to $\frac{2}{3}$ the diameter of the cylinder, and their breadth to $\frac{5}{24}$ of the same.

The diameter and length of the rock-shaft bearing, if subjected to torsion strain, should be from $\frac{1}{3}$ to $\frac{1}{2}$ the diameter of the engine-shaft; if not subjected to torsion, $\frac{1}{4}$ the diameter of the engine-shaft will be sufficient.

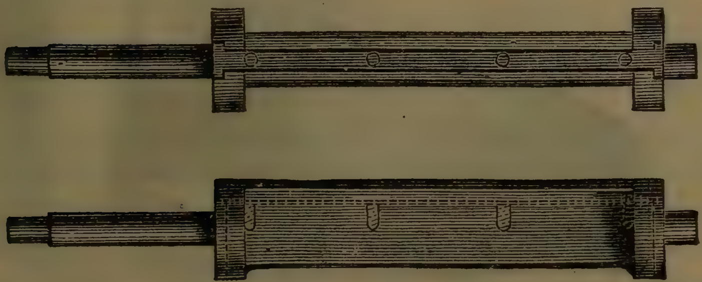
The diameter of the rock-shaft pin should not be less than that of the valve-stem; and if an overhanging pin, it should be from $1\frac{1}{4}$ to $1\frac{1}{2}$ times the diameter of the valve-stem.

In order to make the proportions more plain, it may be advisable to introduce an example; say, for instance, an engine with a cylinder 12 inches in diameter and 30 inches stroke. Thickness of cylinder, $\frac{3}{4}$ inch; depth of piston-ring, 3 inches; diameter of piston-rod, $1\frac{1}{16}$ inches; diameter of crank-shaft, if wrought-iron, 4·8 to 5 inches, if cast-iron, 8 to $8\frac{1}{2}$ inches; length of bearing, $7\frac{1}{2}$ inches; diameter of crank-pin, 2·4 to 3 inches; length of crank-pin, 3·3 to 4 inches; diameter of connecting-rod in the neck, $1\frac{1}{16}$ inches; diameter of eccentric-rod, $1\frac{1}{4}$ inches; diameter of

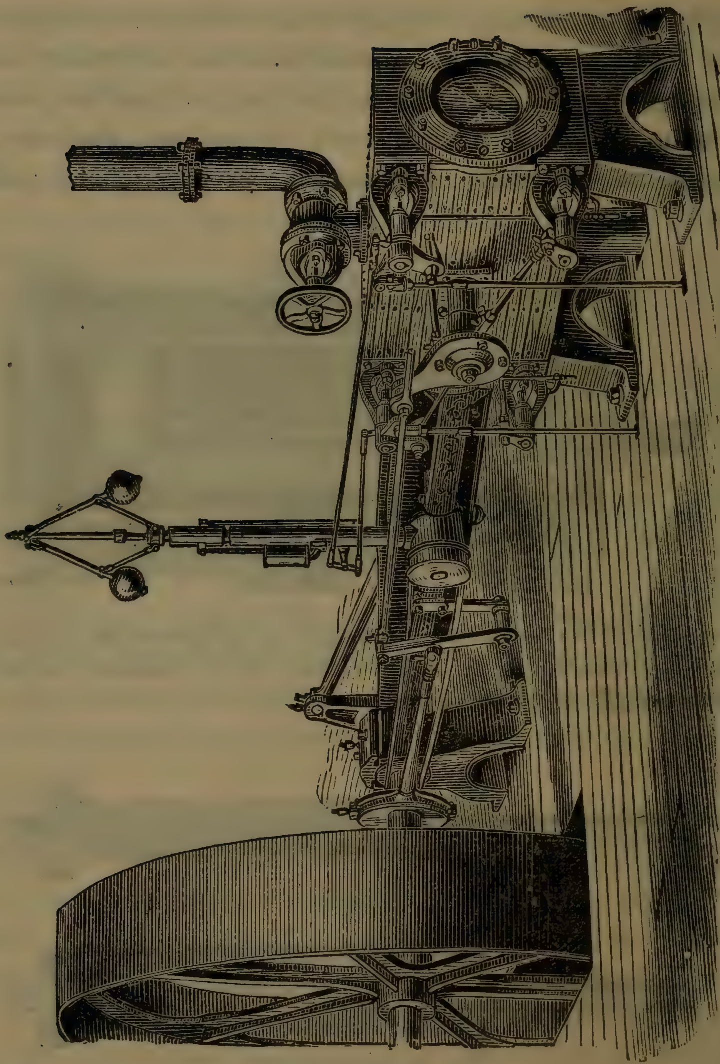
valve-rod, 1 inch; diameter of wrist-pin, 2·4 to 3 inches; length of wrist-pin from 3·3 to 4 inches; thickness of stub-straps, $1\frac{3}{8}$ inches; breadth of straps, $3\frac{3}{8}$ inches; distance from slot to end, 1·8; breadth of gib and key, $3\frac{7}{8}$ inches; thickness of gib and key, $\frac{3}{4}$ inch; clearance, 2 inches to ·218 inch; from key-slot to end of block, $1\frac{3}{8}$ inches; area of steam-port, $7\frac{1}{2}$ inches; length of port, 7·2 inches; width, 1 inch; width of exhaust-port, $1\frac{1}{2}$ inches; diameter of steam-pipe, 3 inches; diameter of exhaust-pipe, $3\frac{1}{2}$ inches.



Section of a Corliss cylinder showing the piston-valves, steam- and exhaust-chests. A, B, Steam-valves; C, D, Exhaust-valves; E, Steam-pipe; F, Exhaust-pipe.



Corliss Valves.



THE CORLISS HIGH-PRESSURE CUT-OFF ENGINE.

SETTING UP ENGINES.

For setting up engines, like setting valves and putting engines in line, no special instructions can be given, as the circumstances of design, construction, location, etc., vary so much; and also the fact that it is rare to find two men agreeing on the best way of performing this kind of work; almost every engineer having his own peculiar way of doing it; yet it may be possible to give general instructions on this point.

Having decided on the location where the engine is intended to stand, line down from the side of the main shaft, if there be any, to the floor at three or four different places in its length. If there be no shafting, measure from the side of the building to the centre, at five or six points in its length; then strike a line across all these points. This line will show with sufficient accuracy the line of the main shaft, or the line of the building, as the case may be. From this line then measure to the point at which the excavation is to be made; and after the earth is removed to a sufficient depth, set up a wooden tamplet on four props, in which to hang the foundation bolts, this tamplet being an exact counterpart of the bottom or under side of the bedplate.

Now drop a plumb-line from two points on the line of the main shaft or the centre line of the building, and from this plumb-line measure to the tamplet, in order to ascertain if the latter is square with the line of the building or the shaft, as the case may be. Then, after the foundation is raised to the proper height, and the bed-plate placed in its position and levelled, draw a line through the centre of the cylinder, and intersect this line with another passing through the centre of the main pillow-block on the bedplate. This latter line will give the exact location of the

off pillow-block, as the axis of the shaft must be at right angles to the horizontal line passing through the centre of the cylinder, and also at right angles to a vertical line.

The fly-wheel may now be swung between the pillow-blocks with a block and tackle, and the shaft slipped through it; the caps of the pillow-blocks screwed down, the piston, cross-head, connecting-rod, eccentric- and valve-rods adjusted, and all the minor details finished up.

DEAD CENTRE.

A difficulty is often experienced in finding what is called the "dead centre," or the position of the crank corresponding to the end of the stroke; although the experienced engineer can in a majority of cases tell by his eye, yet in others, in consequence of peculiarity of design and complication of parts, he finds it very difficult.

A very accurate way of finding the dead centre in horizontal engines, is to place a spirit-level on the top or bottom of the strap of the connecting-rod, and move the crank up or down until the centre is found. Or, if this should be found inconvenient or impracticable, a circle may be described with a pair of dividers on the centre of the crank-shaft equal in diameter to the shoulder of the crank-pin; then place the spirit-level parallel with the shoulder of the crank-pin and the outer edge of the circle; then by moving the crank up or down, as the case may require, the centre can be accurately found. The centres of vertical and beam engines can be found by means of a plumb-line.

HOW TO PUT AN ENGINE IN LINE.

An engine is in line when the axis of the cylinder and the axis of the piston-rod in all positions are in one and the same straight line. This line extended should

intersect the axis of the engine-shaft, and be at right angles to it. The guides should also be parallel thereto. The axis of the shaft must be level, but the centre line of the cylinder may be level, inclined, or vertical, according to the kind of engine.

Take off the cylinder-heads and remove the piston-rod, cross-head, and connecting-rod; then extend a fine line, as nearly as may be by the ordinary means of measurement, through the centre of the cylinder, and let it pass beyond the crank-pin when at outer centre; also let it extend outside the rear end of the cylinder and firmly secure each end to some fixed object at these extreme points. Stretch this line as tightly as it will bear without breaking, and then begin to get it in exact central position by rod measurements.

Mark four points with a centre-punch at equal distances from each other around the bore of the cylinder, say, top and bottom and on each side at each end, and then, by a trial with a small pointed piece of hard wood, set the line so that when one end of the wire rests on each of the four points successively, the other end will just feel the line; next see where this line passes the centre of the shaft. If they coincide, then the cylinder is in line with the shaft; if not, they must be put in line with each other. It is often difficult to move cylinders and shafts, and as one or the other must be removed, in some cases both, in this event only skill and judgment can decide which to do and how to do it.

No special directions can be given how to move the cylinder and shaft into line with each other, because they are so differently constructed; but trusting to the skill of the engineer to secure these two points, the next thing to do is to set the shaft at right angles to the line, and to make it level, also. To do these, turn the shaft until the crank-

pin almost touches the line; and then find, by a rod or inside calipers, if the line lies evenly between the two collars of the pin; if not, note the distance from either one, and then turn the shaft until the pin almost touches the line on the other side, and apply the measure to corresponding places on the collars of the pin. The difference in the measure, if any, will show which way the end of the shaft must be moved to make these measures equal.

The exchanging of the crank-pin from side to side may have to be repeated several times and remeasured, and the shaft moved, before these measures can be made equal. The shaft may require moving endwise in order to get the line to lie evenly between the two collars; but when the turning of the shaft half around brings both collars of the pin the same distance from the line, the shaft is then at right angles to the centre line of the cylinder.

In order to level the shaft by the same method, drop a plumb-line, passing by the centre of the shaft and by the centre line of the cylinder, also; then by turning the pin up and down to the near touching-points of the plumb-line, and raising or lowering the outer end of the shaft until the collar on the pin is the same distance from the plumb-line in both positions, the shaft may be said to be level, or, which is the same thing, it is made at right angles to a plumb-line.

It remains now to bring the guides into line with the cylinder, and this may be done by direct measurement from each end of each guide (if there are two) to the line, moving them until they are parallel to the line and to one another. A spirit-level may be placed on their top faces to show how to adjust them to the horizontal; if no level is at hand, then a true square and plumb-line may be used; and if not, straight-edges placed across the guides, and measurements made down to the centre line, will determine the line of them.

HOW TO REVERSE AN ENGINE.

Place the crank on the dead-centre, and remove the bonnet of the steam-chest; observe the amount of lead or opening that the valve has on the steam end; then loosen the eccentric, and turn it round on the shaft, in the direction in which it is intended the engine should run, until the valve has the same amount of lead on the other end. To determine whether the lead is exactly the same at both ends, a small piece of pine wood may be tapered in the shape of a wedge, and inserted in the port; the marks left on it by the edge of the port and the lip of the valve will show how far it has entered. The engine should then be turned on the other centre, for the purpose of equalizing the lead; the crank should also be placed at half-stroke, top and bottom, for the purpose of determining whether the port-opening is the same in both positions. When the crank is at half-stroke, the centre of the crank-pin is plumb with the centre of the crank-shaft.

SETTING VALVES.

It may seem strange that any person claiming to be an engineer should be found unable to properly set the valves of a steam-engine; and yet it is a fact, that there are thousands of persons having charge of engines, who are unfit, by want of practical knowledge, to do so correctly.

This may arise from the fact that in none of the works heretofore written on the steam-engine, does there appear to be any accurate method laid down for the proper setting of valves; an omission which it is difficult to account for, as it must be admitted that the setting of the valves of steam-engines is among the most important duties the engineer has to perform, involving, as it does, nicety of calculation and mechanical accuracy.

A slide-valve may be properly designed and constructed, and yet be unable to perform any of its proper functions, in consequence of being improperly set, as the steam may be admitted too soon or too late, and the exhaust fail to open and close at the right time, in consequence of which the useful effect of the steam is lost and the power of the engine diminished.

HOW TO SET A SLIDE-VALVE.

Place the crank at 180°, or dead centre, and the eccentric at 90°, or at right angles with the crank; now adjust the eccentric-rod so that the rocker will stand in a perpendicular position; next place the valve centrally over the ports as shown in Fig. 1, and get it equally divided on the

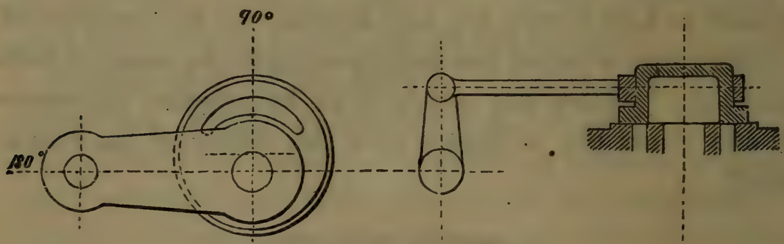


Fig. 1.

rod, so that with the motion of the engine, when all is connected, the valve will travel equally to either extremes from its central position. Then turn the eccentric forward on the shaft, in the direction in which the engine is intended to run, until the valve shows the steam-port just beginning to open, as in Fig. 2. If more lead be required, move the

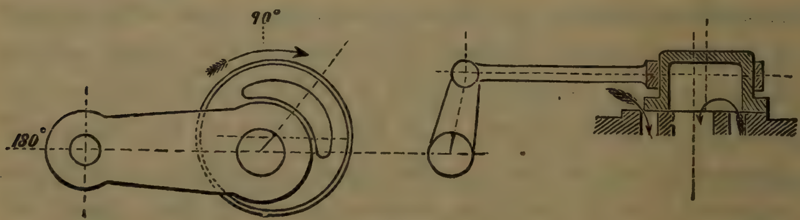


Fig. 2.

eccentric farther ahead, until the valve opens the port to the amount of lead required, when it will be found that, if the valve and ports have been laid out according to the proportions on page 155, the engine will work well. If lap be necessary, it will have to be added to the length of the valve, by either piecing the valve out at each end, or making a new one.

All valves of steam-engines, whether slide, conical, or vibratory, are set in precisely the same way, as the crank must occupy the same positions when the steam and exhaust-valves commence to open, regardless of the design or construction of the engine or valves.

To set poppet or conical valves, it is only necessary to place the crank on the dead centre, and move the cams on the cam-bar until the steam and exhaust-valves have the necessary amount of lead — the throw of the cams to give the required lift of the valves being previously determined, the movements of the cam-bar, the lifts of the valves, the speed, etc., being influenced by the action of the governor in stationary engines. But it must be understood that every different valve requires different setting; a change of speed will necessitate a change in the position of the valve; if the throw of the valve be altered, its lead must also be altered.

One of the best helps to correct valve-setting is a good indicator, as there is nothing known which shows the action of the steam in the cylinder so correctly as this instrument. It tells exactly where the steam goes in and out of a cylinder, because it maps down the motions of the steam as determined by the motions of the valve and piston, recording faithfully the times and the pressures as they actually are, which may be very different from presumed times and pressures as shown by the mechanical movements of the valve and gear.

The valves of all engines, particularly those subjected to high temperatures, such as portable fire-engines, and in fact all engines attached to boilers, should be set when all the parts are warmed up to or nearly to the working temperature, as if valves are set when all the connections are cold, in consequence of the expansion they undergo when exposed to high temperatures, they are liable to travel unequally on their seats or give unequal openings. All corrections, shown by the indicator necessary to be made in the motion of valves, should be done while the parts are warm.

SETTING OUT PISTON PACKING.

Among the most important duties an engineer has to perform, is that of the setting out of piston packing, and as, like many other details of the steam-engine, no general instructions can be given for its adjustment, therefore a good deal depends on the capability and intelligence of the engineer.

The first thing to be done, in order to properly adjust the packing, is to see that the piston is exactly in the centre of the cylinder. This can be ascertained by measuring, with a pair of inside calipers, from the centre of the piston to the inside of the cylinder at four different points. To insure accuracy, calipers with a long and a short leg are generally used, the short leg being inserted in the centre of the piston-head and the circle of the cylinder inscribed with the long one, which will show precisely the position of the piston in the cylinder. The rings should then be set out sufficiently tight to form a steam-tight joint with the inside of the cylinder, and no more. If the cylinder is true and in good condition, the springs of the proper tension, and the rings well proportioned and fitted, there is no reason why the piston should leak.

Whether the piston is leaky or not can be ascertained by removing the back-head of the cylinder and admitting steam to the other end. To make such a trial, the crank should be placed at half-stroke, as many pistons perfectly steam-tight at either end would leak when at half-stroke, in consequence of the cylinder being worn larger in the middle than at the ends.

Brass or composition rings should be adjusted while the cylinder is warm, as, if set out when cold, in consequence of their great limit of expansion when heated, they become too tight, and generate a great amount of friction.

In many instances, where engines fail to develop the necessary amount of power, it is attributed to the leaky condition of the piston, and, as a remedy, the rings are set out to such an extent that, instead of the power of the engine being increased, it is materially diminished, thus aggravating the evil that was sought to be removed.

PISTON- AND VALE-ROD PACKING.

There is probably no part of the steam-engine more frequently out of order, or gives greater annoyance to the engineer, than the piston- and valve-rod packing. Consequently, there is at present, as there always has been since the advent of the steam-engine, a great need of a permanent and reliable piston-rod packing. Such an article could not fail to amply remunerate the inventor,

as it would not only lessen the labor of the engineer, but be productive of very economical results to the owners of steam-engines.

A vast deal of study and ingenuity have been devoted to the removal of this annoyance, and the production of a durable packing, but so far without any very satisfactory results. Wire gauze, gum, soapstone, jute, asbestos, and a great variety of other materials, have been tried, but have failed to meet the requirements of a permanent piston-rod packing.

In the early days of the steam-engine, hemp was the material most generally used, and answered very well, as it had the advantage of being always ready for use, and requiring no special tools, particular size of stuffing-box, or extra skill to use it; but its usefulness had a very narrow limit, particularly where steam of a high pressure was used, as it soon lost its elasticity, and consequently became worthless.

Soapstone gives tolerably good satisfaction, as, when first put in, it has the advantage of producing less friction than any other material used for the purpose; but it has the disadvantage of becoming hard, which is very objectionable; as, when packing loses its spring or elasticity, it greatly interferes with the smooth and easy working of the engine, particularly in the case of those out of line.

The failure of any of the different kinds of packing now in use to give satisfaction might be attributed in part to a want of depth in the stuffing-boxes of modern built engines, as, when only a small quantity can be used at a time, the stuffing-boxes have, of necessity, to be screwed up very tight, which has the effect of producing extra friction, which soon renders it worthless.

Engines in general are packed less frequently than they should be; but this is a mistake, as all that is saved in

packing is frequently lost by the fluting of the rods, when the packing becomes hard and dry.

It is always better to have packing sufficiently large to admit of flattening before being inserted in the boxes.

Before packing the piston- or valve-rods, all the old packing should be removed, also all dust or dirt that may have accumulated in the stuffing-boxes. The rings should be inserted in opposite directions, so as to break joints; but the ends of the rings should not be permitted to meet, as this has the effect of preventing them from hugging the rod when they stretch. After the packing has been screwed down into the bottom of the box, the nut or nuts should be unscrewed one or two threads, in order to allow it to expand.

Old files or other rough instruments should never be used for removing packing from the boxes, as they have a tendency to scratch the rods; but every engineer should provide himself with a smooth steel packing-bar and packing-hook to use for that purpose.

If, after an engine is packed, the leakage should continue excessively around the rods, it is always better to remove one or two pieces of the packing, and replace them again, than to continue screwing up the stuffing-box. When it becomes necessary to tighten up the packing, it will be found of great advantage to do so when the engine is standing still.

Piston- and valve-rod packing should always be kept in a clean place, and out of the reach of dust, sand, ashes, or any substance that would be likely to cut or flute the rods.

The proper size of packing for any rod is one-half the difference between the diameter of the stuffing-box and the diameter of the rod.

If a stuffing-box is extra large, and the quantity of

packing on hand be insufficient, one ring of clean lead pipe inserted in the bottom of the box answers a very good purpose.

AUTOMATIC CUT-OFFS.

Variable cut-off engines are engines having their steam-valves so controlled by the governor as to promptly cut off the steam at any point from zero to half-stroke; the cut-off taking place earlier or later to accommodate the varied loads on the engine and varied pressures in the boiler,—the object being to obtain full boiler pressure at the commencement of each stroke, and maintain it to the point of cut-off, leaving the balance of the stroke to be completed by expansion,—the speed of the engine being controlled by the cut-off, and not by throttling.

Until quite recently, the common method of regulating the flow of steam from the boiler to the cylinder has been by the throttle-valve—a kind of "damper"—in the steam-pipe, which was turned as the speed of the engine increased, and choked off the supply of steam,—or the steam, in its passage from the boiler to the cylinder, had to ooze through the contracted crevices of some peculiar type of governor-valve. An engine controlled by any such devices is in a condition somewhat like that of a horse restrained by a brake applied to the wheels, and compelled to exert more strength than is necessary. These relics of barbarism are fast giving place to the system referred to in the foregoing paragraph, which removes the brakes from the wheels and puts the bit in the horse's mouth instead.

Although all intelligent engineers are agreed upon the superior economy of the automatic cut-off engine, few—excepting those who have had the opportunity of making a practical comparison—are aware of the great saving in

the expense of fuel, over that class of engines wherein the point of cut-off is invariable relative to stroke of piston. It is quite well understood, that the amount of work realized, as compared with the total theoretical work due the volume of steam expended, even in the most perfect engines, as shown on page 70, is a very small percentage of the whole energy; and it is, therefore, the more an object of interest to know precisely what the difference is between these two classes of engines in point of economy.

In engines with a variable expansion-gear controlled by the regulator, there is no impediment (save such as may occur at the port entrance) to the free flow of steam from the boiler to the cylinder; the regulation being effected, not by diminishing the pressure, but by cutting off in the cylinder the volume of steam necessary for each particular stroke; consequently, the only loss in pressure between the boiler and cylinder is that due to the length and number of bends in the conducting-pipe. Whilst even in the best throttling engines, in consequence of the peculiar construction of the governor-valve and the tortuous passages through which the steam is forced to travel, the pressure in the cylinder is, in a majority of cases, reduced from three-fourths to one-half that existing in the boiler; the evil effects of which are shown on page 63.

It may seem strange that any intelligent engineer or steam-engine builder should deny the superiority of cut-off over throttling engines; and yet there are some who argue, that the economy in the use of the cut-off engine lies more in the representations than in the excellent performances,—which, of course, is an unpardonable error.

In the opinion of the writer, the conditions of admission and suppression of steam to the cylinder insuring the highest grade of economy, are a full port with no intervening obstructions to impede the free flow of the steam,

and a rapid movement of the cut-off or steam-valve over the port; as mere increase in the mean effective pressure, resulting from a tardy closing of the port, represents no gain during one stroke of the piston, that may be stored up and expended during the succeeding stroke; hence, any force upon the piston in excess of that required to balance the resistance, will result in a diminished economy.

The economy of a high-pressure steam-engine is exactly in proportion as its average piston pressure is higher than its pressure when it exhausts, provided the pressure shall not fall below that of the atmosphere; the highest economy being attained when the stroke is commenced with full boiler pressure, and the steam quickly and completely cut off at a point in the stroke that allows the pressure to fall to or very near that of the atmosphere; the full boiler pressure to be maintained from commencement of stroke to point of cut-off.

GOVERNORS.

The use of the governor is to preserve a perfectly regular speed in the engine, by varying the supply of steam as the work of the engine varies, which is an object of paramount importance in the prosecution of many manufacturing purposes, particularly in cases where, in consequence of the peculiar character of the work, it is found impossible to confine the variation in power within narrow limits.

Centrifugal force has received the most attention, and nearly all governors since the days of Watt have been constructed on that principle; consequently, hundreds of patents have been issued to inventors in which it has been attempted to combine sensitiveness of action and strength; but the problem still remains unsolved, as all ball gov-

ernors have the defects of requiring heavy balls, and of demanding a wide range of action where they have considerable force to overcome.

It is well known that a governor that is very sensitive cannot be very powerful, nor can one that is very powerful be very sensitive; and that, in order to obtain great power from the ordinary ball governor, it is necessary to use very heavy balls or springs, or a very high speed, on the principle that a great resistance requires a great controlling force.

The cut on page 142 represents the Huntoon Governor, in which it will be noticed that the fly-balls are dispensed with, and the principle of the screw propeller adopted, and which is claimed to be the only isochronous* governor in this country.

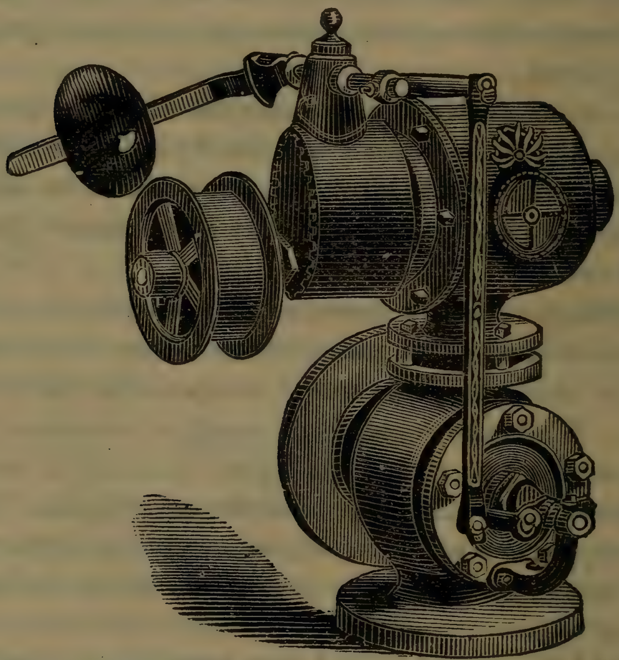
In this governor, a screw, rapidly rotating in a closed tank containing oil or water, exerts a force in the line of its axis, which is made use of in operating the throttle-valve. While the engine is at speed, no movement of the valve occurs, but should the speed diminish, a weighted arm forces back the screw and the valve opens. It will continue to open until the engine comes up to the proper speed again, whatever the conditions as to the load or steam pressure.

Should the speed exceed that intended, the screw acts more energetically upon the liquid in which it works, and the increased effort is sufficient to overcome the resistance of the weighted arm, and to close the valve until the proper speed is again acquired.

All marine engines, but especially screw engines, are liable to sudden fluctuations of speed from the varying immersion of the propeller in a rough sea; and the necessity of employing some species of governor to redress these irregularities has long been perceived.

* Uniform in action and time.

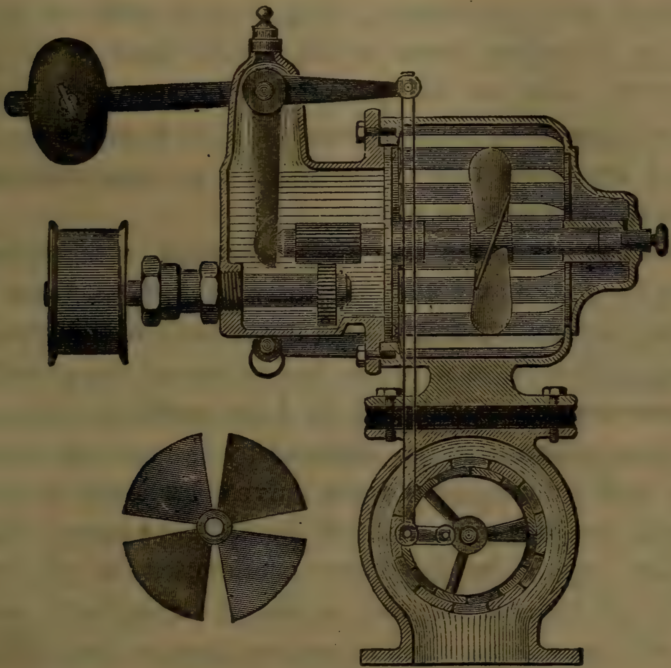
The common form of governor that is used on land engines is obviously inapplicable to such a purpose, as the balls would open and close by the heaving of the ship. Various kinds of marine governors have been tried, and in turn abandoned; and whether it is possible to construct a governor that would control the speed of marine engines under all the varying conditions to which screw-



The Huntoon Governor.

propellers and paddle-wheels are subjected, in all kinds of weather, remains yet to be proved, as none of the marine governors heretofore tried, either in this country or Europe, have given sufficient satisfaction to warrant their adoption into general use. In steamships, the racing in heavy weather, when the screw is at times but partially submerged, becomes a very dangerous matter, and frequently causes serious breakages; though, in rolling mills,

the variation in horse-power is not so great as in steamers, because the total horse-power is not as large. Yet, when the engine has to exert all its power to force a piece of iron or steel through the rolls, and is then suddenly relieved of everything except the friction of the machinery, almost any of the ordinary forms of governors are helpless in such cases. The Huntoon Governor has been ap-



Interior View of the Working Parts of the Huntoon Governor.

plied to marine engines, and is said to have answered all the requirements of a good marine governor. It is very desirable for marine purposes on account of its simplicity of design, fewness of parts, and compactness of form. It is very extensively used on stationary engines, with the most satisfactory results.

The governor has received a good deal of attention within the past few years from engineers and inventors,

and the result has been the production of a good many excellent regulators for controlling the speed of throttling engines, among which might be mentioned Judson's, Conde's "American," Shive's, Gardener's, Pickering's, Brown's, Jenken's, etc.; but what is still needed, is a governor which will compel an engine to "come to speed" under all variations of steam pressure and of load. Such governors are very scarce; and, if they exist at all, it seems to be rather more in the claims than in the performances. But as the production of such a governor does not appear to be insurmountable, it is probable that some of the forms now in use will be so improved as to accomplish the desired object.

Governor - spindles working through stuffing-boxes should be frequently and carefully packed, as, when the packing becomes old and dry, if screwed up to prevent leakage, it interferes with the free action of the governor.

Governors should always be kept perfectly clean and free from accumulations induced by the use of inferior oil, as such gummy substances have a tendency to interfere with the easy movement of the different parts. Many first-class regulators have been condemned as not being capable of controlling the engine at a uniform speed, when all that was required to restore them to their original accuracy was simply a good cleaning.

Rules for Calculating the Size of Pulleys for Governors.—*To find the Diameter of Governor Shaft-pulley.*—Multiply the number of revolutions of the engine by the diameter of the engine shaft-pulley, and divide the product by the number of revolutions of the governor.

To find the Diameter of the Engine Shaft-pulley.—Multiply the number of revolutions of the governor by the diameter of the governor shaft-pulley, and divide the product by the number of revolutions of the engine.

THE CATARACT.

The cataract supplies the place of the governor in the single-acting Cornish pumping-engines. It consists of a small pump-plunger and valve, set up in a cistern of cold water, the valve opening inwards, so that when the plunger ascends, the water passes through, it being supplied by a cock, which is worked by means of a rod from the beam overhead. If the cock be shut, the plunger cannot descend; but if slightly opened, it will descend gradually; and as soon as a certain quantity of water has passed through, its weight opens the injection-valve and condensation takes place, when the engine can complete its stroke; for the engine can only make the stroke as the water is supplied for condensation.

In this manner, the cataract regulates the speed of the engine; for if the cock be fully open, condensation takes place at once, and if only partly open, condensation will be delayed until the water is supplied.

There are other varieties of cataract employed besides that here described, but they all depend upon the same general principle. In some cases air is used instead of water, and in others a cylinder of oil is employed, fitted with a piston with a valve, after the manner of a pump-bucket and a small side-pipe, fitted with a cock, which communicates between the spaces on each side of the piston.

When the piston of this cataract is forced down, the oil easily ascends through the valve into the upper part of the cylinder; but when it is drawn up, the oil can only escape by the curved pipe, from the space above the piston to the space beneath it, by passing through the contracted orifice of the cock; and, though a considerable counter-balance be applied, the piston, if the cock be partially closed, can ascend but slowly. The effect is the same as in the arrangement first described.

WRIGHT'S HIGH-PRESSURE ENGINE.*

This engine is built upon a solid, square, cast-iron bed-plate. The cylinder is fitted with Wright's patent slide-valve cut-off, worked by steel cams upon a horizontal shaft, to which a longitudinal movement is given by the governor.

The cylinder is connected to the main pillow-block bearing by a strong wrought-iron brace rod. About the centre of this rod is located a cast-iron standard to steady the rod, and also to furnish a bearing for the rock-shaft. The governor, it will be noticed, is enclosed in a brass shell or urn, and is located upon the top of this standard, and is driven by a belt from the crank-shaft. The governor-spindle passes down through this standard, and gives a longitudinal movement to the cut-off shaft through levers which vary the cut-off to any point in the stroke necessary to balance the load on the engine.

The cut-off cams upon the side shaft have a rocking motion, derived from an eccentric on the main shaft communicating with a bell-crank, and from this bell-crank through a universal connection with the side shaft. This one eccentric gives the motion not only to the cut-off, but to the exhaust-valves. The concentrating of nearly all the valve-gearing in one place adds not only to the beauty and simplicity of the machine, but is much more convenient to adjust and handle.

An unhooking arrangement is fitted to the pin in the belt-crank, whereby the engine can be stopped and reversed by hand at pleasure.

The cross-head is carried in cast-iron slides, which are solid and cast on the bed. The feature of the cross-head is its stability, which is secured by its having a very broad

* See page 20.

bearing on the bottom and sides, as also a top bearing with brass gibs. These engines have acquired a very wide reputation on account of their excellent performances.

HAWKINS AND DODGE'S HIGH-PRESSURE ENGINE.*

The bed-plate of this engine is cast in one piece, of a depth and form to give it the necessary rigidity. The cylinder has cast on one side a shelf for the valve-seat, with steam-ports cored therein, on which the steam-chest, containing the main and cut-off valves, is bolted.

By taking the nuts off the bolts, the cover, chest, and valves may be easily removed, thus making the valve-seat easy of access, if, from wear or other causes, the valves may need refitting; which, on engines with the steam-chest cast on the side of the cylinder, is often a difficult and unsatisfactory undertaking.

The main valve is an ordinary slide-valve, operated by an eccentric; it is formed with four narrow ports on its back corresponding to the same arrangement of ports in the cut-off valves, which work on the planed surface of its back, they being operated entirely by a Porter governor. By the simple adjustment of the lever on the governor-rod, they may be made to cut-off at any point of the stroke, while the periods of steam inlet and exhaust are unaffected.

The rods are of steel, and all connections straight; the bearings are large and long, rendering them less liable to cut and wear; the different parts, besides being bolted, have taper dowel-pins fitted into reamed holes, so that the whole may be taken apart and put together again in perfect line.

The engine consists of but few parts; it is constructed in

* See page 24.

a compact and symmetrical form, and finished in the best style of workmanship; and is liable to no derangement, and requires but ordinary care to keep it in working order. The cylinder is encased in a neat, polished sheet-brass cover or fluted cast-iron jacket, thus preventing radiation. The reputation of these engines for economy and durability stands deservedly high.

WATTS AND CAMPBELL'S HIGH-PRESSURE ENGINE.*

The most remarkable features of this engine are the valves and valve-gear. There are four valves, two steam and two exhaust, which are operated by cams, the throw of which is controlled by the governor. The expansion-gear is Hornig's cut-off, which opens and closes the valves suddenly, and retains them at rest at full port, opening while the steam is passing in or out.

In consequence of the perfect arrangement of the valve-gear, the steam is admitted and exhausted very freely, which has the effect of preventing the possibility of much back pressure.

The bed-plate is heavy and well proportioned; the cylinder is protected by a metallic jacket, and the whole engine presents a very neat and substantial appearance.

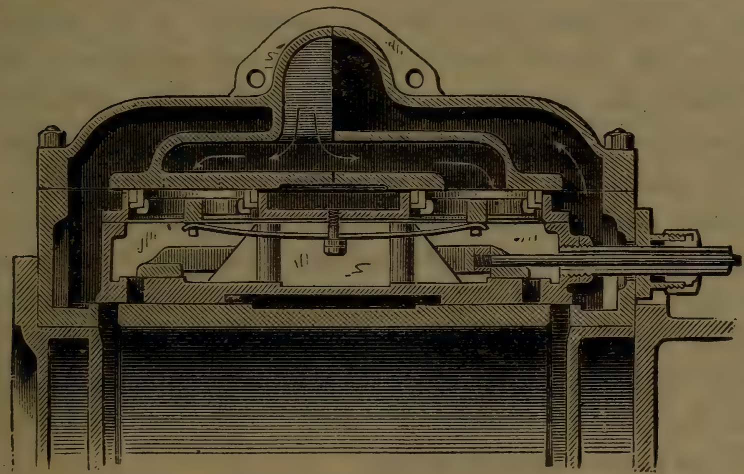
THE BUCKEYE HIGH-PRESSURE ENGINE.

The cuts on pages 52, 53, represent the Buckeye automatic cut-off engine, which is claimed to satisfy all the conditions necessary for the highest attainable economy in the use of steam-power. The steam is admitted to the cylinder through the slide-valve (a cut of which is shown on the opposite page) instead of around it. It enters the valve through circular openings in its back, and is

* See page 37.

admitted from thence to the cylinder through two ports, which are alternately brought to coincide with the cylinder-ports.

To the openings in the back of the valve are fitted steam metal self-packing rings, which serve the purpose of insuring a steam-tight connection between the interior of the valve and the live-steam chamber in the back of the chest. The area of these openings is made just sufficient to hold the valve to its seat; hence it is as nearly balanced as is practicable or desirable. As the valve-chest contains only exhaust steam, the engines may be run with the chest-lid removed, and any leakage readily detected.



The exhaust takes place at the ends of the valve into the ends of the chest, thence through ample passages into the exhaust-pipe, passing downwards, as shown in cut on page 52, thus avoiding the tortuous exhaust passage involved in the use of the common slide-valve.

The cut-off valve works inside of the main valve, and alternately closes the ports leading to the cylinder. The fixed eccentric operates the main valve, and an adjustable

one operates the cut-off valve through the medium of a small rock-shaft, which works in a bearing in the rock-arm of the main-valve gear, and moves with it. The movement of the cut-off valve, relatively to its seat in the main valve, is thus, both as to time and extent, just what its eccentric would produce if the valve worked in a stationary seat, and was attached directly to said eccentric. The eccentric-rod of the main-valve gear works horizontally, while the cut-off eccentric-rod inclines downward, so that its attachment to its rocker-arm may be on a level, or nearly so, with the centre line of the main rock-shaft. This arrangement is shown in the cut on page 52.

The stem of the cut-off valve passes through the hollow stem of the main valve, and is connected to an upright arm on the cut-off rock-shaft, on the end opposite to that to which the eccentric-rod is attached. The automatic adjustment of the cut-off eccentric is effected by means of its connection with two weighted levers contained in a circular case on the engine-shaft. The outward movements of these levers advance the eccentric forward on the shaft, and two well-tempered cast-steel wire-coil springs furnish the centripetal force which returns them when the speed slackens. The effect of this arrangement is such that the varying changes of speed due to load and pressure, are almost imperceptible.

All the wearing parts of these engines are made of the best material, and are proportionally equal to those of the most approved class of engines. The piston- and valve-rods, wrist- and crank-pins are made of cast-steel, and the connecting-rod and pillow-block boxes are of best machine brass and Babbit-metal.

WHEELOCK'S HIGH-PRESSURE ENGINE.*

The bed-plate of this engine is of a pattern in very common use, and forms a butt-joint with the cylinder like the Corliss and Allen types of engine. The steam is admitted and exhausted by a single valve at each end of the cylinder.

The valves, like those of the Corliss engine, are suspended in sleeves or bushings, and are located at each end and directly under the cylinder, by which arrangement the clearance is reduced to a very small amount.

The valve-gear is constructed in the simplest form of the ordinary eccentric-rod and wrist-pin, and can be at any time disengaged without difficulty. Motion is imparted to the valves by cranks keyed into the stems, similar to those on the Corliss valves, which, being securely fastened, makes any derangement of the valves inside impossible; while the uniformity of wear is secured by the surfaces passing entirely over each other at every revolution of the engine. By a peculiar arrangement of screw and check-nut, which form a joint or shoulder on the valve-stem, the necessity of stuffing-boxes is entirely obviated.

The steam-chest being located underneath, and forming a part of the cylinder, serves as a reservoir, in which the drips are located, thus guarding against any injury that might otherwise arise from the accumulation of water in the cylinder. The great drawback of these engines is the impossibility of making the valves steam-tight, in consequence of their peculiar form (cylindrical), which prevents the use of the scraper; hence, their liability to become leaky.

THE HARRIS CORLISS HIGH-PRESSURE ENGINE.†

This engine has a massive bed-plate, which rests on cast-iron legs bolted to the pillow-block and cylinder, one end

* See page 71. † See page 126.

of which forms the front head of the latter. They have also large fly-wheels, which serve the double purpose of a balance-wheel and driving-pulley.

There are four valves, two steam and two exhaust, placed at the extreme ends and directly upon the bore of the cylinder; being made independently adjustable, it follows that the time of commencement, extent, and rapidity of the movement of each is capable of being arranged accurately.

Motion is imparted to the valves by a single eccentric acting through the medium of a vibrating disk, sometimes called a wrist-plate, from which the valve connections radiate. Apart from the simplicity of this device, an important advantage is gained by the utilization of the crank motion's known irregularity to give the valves a rapid motion at the instant of opening or closing.

The steam-valves are placed on top of the cylinder and open directly into the clearance, therefore there are no long passages to fill with steam at each end of the cylinder. The same valve admits and cuts off the supply of steam, no auxiliary valve being necessary.

The exhaust-valves are situated under the cylinder, at the clearance spaces, and can therefore free the cylinder of water, without the use of other devices, in the most thorough manner. They, like the steam-valves, are situated close to the bore of the cylinder, and therefore have no long passages to fill with live steam.

The steam-valve commences to open its port at one end of the cylinder when the eccentric is producing its most rapid movement, and, as the motion of the eccentric is declining towards the end of the throw, an increasing speed is obtained by means of the wrist-plate, which compensates for the slow motion of the eccentric. At the same time, the steam-valve at the opposite end of the cylinder commences to *lap* its port, by the motion of the eccentric,

but by a reverse or subtraction of speed, produced by the same wrist-plate, which speed is constantly decreasing till the throw of the eccentric is completed. The lapping and opening of the steam-ports require each the same amount of throw of eccentric, producing, say, for instance, a lap of half an inch at one end of the cylinder, while the opposite end would have an opening of more than twice that amount.

The exhaust-valves are moved by the same eccentric and the same wrist-plate before spoken of; but they have a much greater travel for the purpose of giving the engine a free exhaust. The exhaust-ports are as long and twice as wide as the steam-ports.

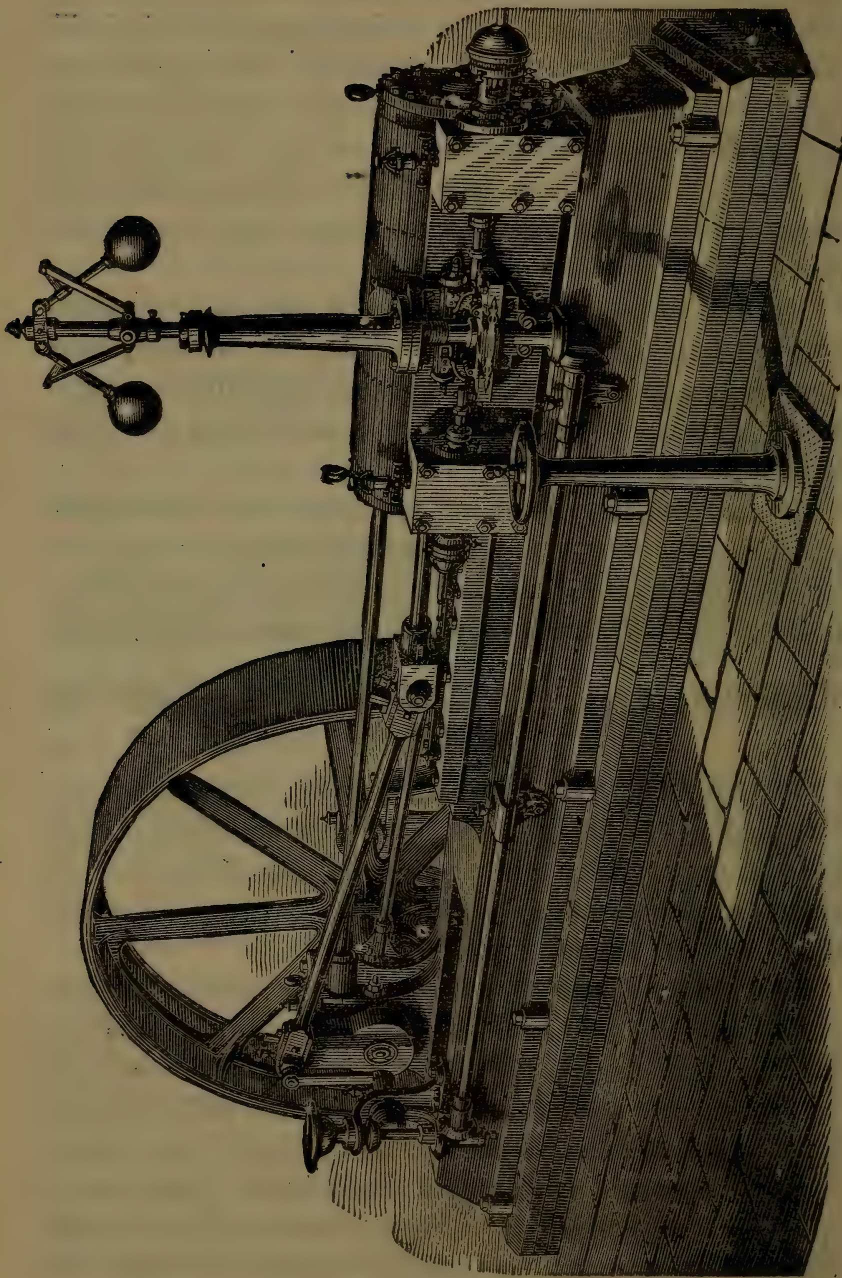
At the commencement of each stroke, the full boiler pressure enters the cylinder, and the motion of the governor determines at each half revolution where the steam is to be cut off, that the proper speed may be maintained.

HAMPSON AND WHITEHILL'S HIGH-PRESSURE ENGINE.*

The cylinder of this engine is securely bolted to a heavy bed-plate, and the supports for the guides to the cross-head and the main pillow-block are cast on the bed. A stout wrought-iron brace is placed between the main pillow-block and the exhaust-valve chest.

The steam-chests, as shown in the cut, are bolted to the cylinder on one side and the exhaust-chests on the other; the steam-valves are horizontal poppet, and are operated by cams on the governor-spindle; while the exhaust-valves are plain slide, double-ported, and receive their motion from an eccentric on the main shaft, which gives them a quick and large opening. They are connected to the valve-stems in such a manner that the wear on the face and

* See page 154.



HAMPSON AND WHITEHILL'S HIGH-PRESSURE CUT-OFF ENGINE.

seats is taken up by the steam acting on the backs of the valves. The lower edges of the exhaust-ports are below the line of the bore of the cylinder, which facilitates the escape of the water of condensation from the cylinder and exhaust-chests.

The governor is of the ordinary centrifugal, ball kind, and is claimed to be more powerful, less liable to get out of order, and to control the speed of this class of engines, better than any other form now in use. It receives a positive motion from a pair of mitre-gears on a horizontal shaft connected with the main shaft of the engine.

The valves are readily accessible by simply removing the bonnets of the steam- and exhaust-chests, and, as all parts of the valve-gear are exposed to view, any derangement can be easily detected. The expansion-gear is what is known as Penny's automatic cut-off.

A very high economy in the use of steam is attained in these engines.

THE ALLEN HIGH-PRESSURE ENGINE.

The Allen engine (a cut of which is shown on page 161) is based on the principle that work should be performed by the development of high velocity in a small mass; consequently, the fly-wheel is of much less magnitude than is used on engines of equal power.

The main bed-plate extends from the front cylinder-head, to which it is bolted, to a point sufficiently beyond the crank-shaft for receiving the large pillow-block of the same. This bearing is made unusually long, and thus insures absolute rigidity of the shaft.

There are two valve-chests on the side of the cylinder—one for the valves regulating the steam, the other for those controlling the exhaust. The former valves are rectangular, and work between scraped surfaces. They are

balanced by the pressure of the steam on their opposite sides, while the latter are of the same construction as the ordinary flat slide-valves.

All the valves are actuated by one stationary link. The motion is transmitted to those regulating the exhaust through a rocker having opposite arms; the connecting-stem, being pinned rigidly to the top of the link, imparts an invariable action to the valves. The steam-valves have their stems attached to two other rockers, and these in turn have connecting-stems pinned to the sliding-block of the link. To this block the governor is connected, which, acting under an increased or diminished velocity, causes the block to traverse the link, and effects an earlier or later cut-off of the steam.

In consequence of the high speed at which this engine is generally run, which is from 600 to 800 feet travel of piston per minute, it has been found advisable to use a solid piston having some grooves turned in it, in which the water of condensation from the first steam that enters the cylinder lodges, and forms a packing of sufficient resistance to prevent leakage.

The crank- and wrist-pins and the piston- and valve-rods are invariably made of steel; the guides and jaws of the cross-head are faced with the same material, and the boxes are usually made of bronze or gun-metal.

WOODRUFF & BEACH'S HIGH-PRESSURE ENGINE.*

The bed-plate of this engine is of the ordinary form, cast in one piece, and is bolted to the foundation by means of an inside flange. The cylinder is bolted and dowelled to the bed-plate, which prevents the possibility of it being moved from its original position.

* See page 165,

The steam-valves are conical or poppet, and receive a horizontal motion from a cam on the governor-spindle, which regulates the amount of opening according to pressure and speed, the closing of the valves being effected by spiral springs enclosed in sleeves on the ends of the steam-chests. The governor rests on a shelf above the steam-chests, directly in the centre of the cylinder, and receives a positive motion from a horizontal shaft driven by a spur-gear on the main shaft of the engine.

The steam-valves are located at each end of the cylinder, and open directly into the clearance, thus reducing the cubic contents of the steam passages. The exhaust-valve is an ordinary flat slide, situated below the bore of the cylinder, and is operated by a double crank of short throw, which receives its motion from a pair of mitre-gears on the same horizontal shaft that works the governor. This arrangement admits of a free exhaust with a very limited travel of valve, and also allows an easy escape for the water of condensation from the cylinder. This form of exhaust-valve is somewhat objectionable, as the wear is entirely on its back, consequently the pressure of steam against its face has a tendency to force it away from its seat, and induce leakage.

These engines, like most automatic cut-off engines, are capable of developing a great amount of power, as the steam is admitted and exhausted very freely, thus preventing the possibility of much back pressure.

NAYLOR'S VERTICAL HIGH-PRESSURE ENGINE.*

The design of this engine is very creditable, for, while it possesses strength and solidity, there are only three principal parts — the cylinder, frame, and circular base. The slides and the pillow-blocks are cast with the frame, so

* See page 300.

that they cannot become loose, and in consequence of their vertical position, the side wear of the cylinder, cross-head, and guides is very slight.

The piston-rod, crank- and wrist-pins, and valve-stem are made of steel, and all the bearing surfaces are large and well fitted; the adjustment for taking up the wear in the revolving parts is made in the most approved manner. The running gear is accurately balanced, so that a very high piston speed may be attained without any vibration or jar.

The cross-head has adjustable gibs, turned to fit the guides, which are bored out exactly with the line of the cylinder. The cylinder is handsomely lagged with black walnut and banded with brass, and the glands and stuffing-boxes are made of the latter material and nickel-plated. The valve is the ordinary expansion-slide, but Cooper's patent balanced slide-valve is used when required. The speed of the engine is controlled by an improved governor, manufactured at the same works.

WILLIAMS' VERTICAL THREE-CYLINDER HIGH-PRESSURE ENGINE.*

In this engine three cylinders are used, and each cylinder is single-acting, receiving the steam upon the upper side only of the piston. The connecting-rods are attached directly to the pistons, and actuate a three-throw crank-shaft.

Each piston serves as a steam-valve, and controls the supply of steam to one or the other of the two remaining cylinders. There is a steam chamber in each piston, and a port in its side. Steam is supplied from the boiler by means of a tube passing through the top of the cylinder and into a steam-chest.

When the piston has reached about three-fourths of its

*See page 319.

downward stroke, the steam-port in it overlaps a port formed in the side of the cylinder, and steam then passes to the top of another of the cylinders. When, on the other hand, the piston has reached about one-half its return-stroke, it uncovers the port in the side of its cylinder, and allows the steam to escape from the cylinder, into which it was previously admitted, into a casing round the crank-shaft, from which the exhaust-steam is taken either to a condenser or to the air, as the case may be.

The advantages claimed for this class of engines are cheapness, simplicity of construction, fewness of parts, and an almost unlimited speed. But while it may be admitted that the whole arrangement is simple and compact; yet it is difficult to see what advantage can be gained by its use, over that of the double-acting arrangement, as, when the cylinders become worn or the pistons leaky, they involve the expense of reboring three cylinders, or readjusting three pistons; for, unless they are perfectly steam-tight, they must continue to be a great source of annoyance.

ROPER'S CALORIC ENGINE.

The cut on p. 324 gives an elevation of Roper's* caloric engine, showing the valves, valve-gearing piston, cylinder, air-pump, beam, and fly- and band-wheels. The cut on p. 325 shows a vertical section of the same, in which No. 1 represents the piston, on the top of which is fastened a leather packing; 2, the piston-drum, made sufficiently long to keep the packing from the fire; 3, a lining of asbestos filled in between the shell and the sheet-iron lining which surrounds the fire-brick; 4, the brick lining of the fire-box; 5, the outer shell of the engine; 6, an iron ring to fasten the packing to the piston-head.

* The writer is not the inventor.

To start the engine, it is only necessary to turn the fly-wheel half a revolution, in order to give the plunger of the air-pump an upward motion, when the cold air is drawn in at the opening, A; on the return of the plunger, the valve, B, closes, and the air is forced into the engine through the check-valve, D.

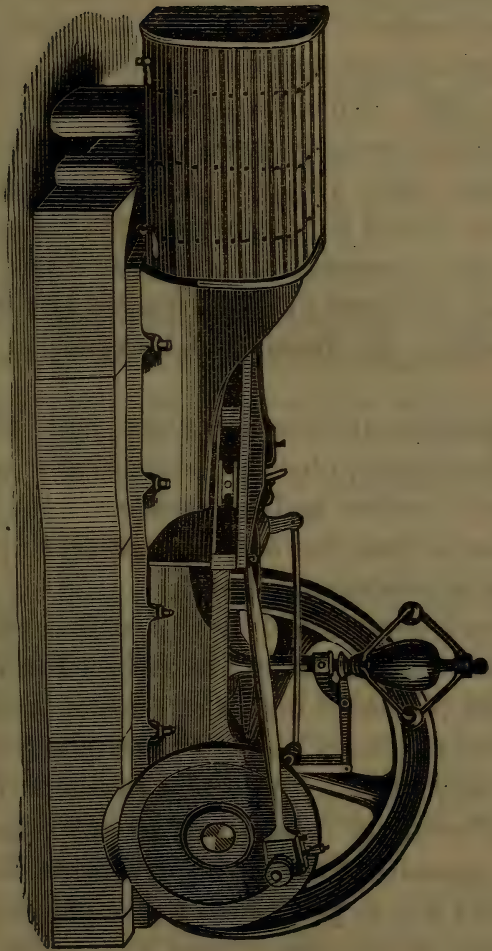
If the lower damper, E, is open, and the upper damper, H, is closed, all the air will enter the fire-chamber under the grate, F, and pass through the fire. If the upper damper, H, is open, and the lower damper, E, is closed, as they should always be after the fire is in good condition, the air all passes over the fire.

The expansion takes place in the fire-box, and, the doors being closed air-tight, the air remains under pressure until, by the eccentric motion communicated to the valve-toes, the inlet poppet-valve is opened, which allows the compressed air to pass into the cylinder, which forces the piston up; its place in the fire-chamber being supplied with cold air at the same instant by the downward movement of the air-pump plunger.

At this moment, by the same motion, the outlet or exhaust-valve is opened, and the inlet-valve closed. The force of the balance-wheel and the weight of the piston bring the piston back; and the expanded air, returning by the same passage, finds the inlet-valve closed and the outlet or exhaust-valve opened, and is discharged through the funnel or chimney.

Many of the mechanical difficulties heretofore experienced in the employment of this class of motors seem to have been successfully overcome in this engine; but still there are natural difficulties which neither chemistry nor mechanical science have been able to remove up to the present time, nor is it at all likely that they ever will be.

The first great drawback to the use of heated air is the



THE ALLEN HIGH-PRESSURE CUT-OFF ENGINE.

small amount of its expansion. Water expands 1700 times; so that to obtain a volume of expansion of 1 cubic foot, it is only necessary to force into the boiler 1 cubic inch of water. Now, could air be obtained in any similarly condensed and manageable form, yet retaining its present small capacity for heat, it would stand on a totally different footing from that which it actually occupies.

Even at 568°, (which is probably above the temperature at which it could be practically used with advantage in a cylinder with air-tight piston,) its volume is only double what it is at 60°; consequently, for every volume of heated and expanded air which develops power as it escapes, half a volume of cold air must be forced into the reservoir where the heating and expansion are accomplished.

This operation at once consumes half the theoretic power of the engine, plus the friction of the supply cylinder with its valves and appendages, and increases its consumption of heat for duty to considerably more than double that of the steam-engine.

The second great disadvantage under which an air-engine labors may be said to be included in the first. It is this: that the small degree of tension it is possible to employ, from the limit placed by the question of temperature, not only virtually precludes the employment of the principle of working expansively to any extent, but also entails the necessity of employing cylinders of an enormous and unwieldy size in proportion to the power obtained.

By cutting off the steam when the piston of a steam-engine has made $\frac{1}{8}$ of the stroke, its duty may be increased more than threefold, reducing the consumption of fuel to 33 per cent.

Now, suppose expansion be carried sufficiently far in the air-engine to reduce its consumption of fuel to 33 per

cent., thus placing it on an equality with the steam-engine, with regard to the consumption of fuel in proportion to the amount of pressure exerted on the piston, the steam-engine would still possess an immense advantage over the air-engine in practical utility and convenience on account of the huge bulk of the latter.

An air-engine capable of developing power equal to that of a steam-engine would require a working cylinder with an area from 20 to 50 times greater than that of the steam cylinder, together with an air-pump of at least two-thirds the area of the working cylinder.

Nevertheless, the caloric engine, in its present improved form, has for some years past been successfully employed as a hoisting-engine in stores and warehouses, and also as a pumping-engine at railway stations, hotels, and country residences. It is also desirable for yachts and agricultural purposes, on account of its simplicity and freedom from explosion.

HASKIN'S VERTICAL HIGH-PRESSURE ENGINE.*

There are a great number of engines of this class throughout the country, and there is no very material difference in their general appearance. Such engines are selected more for economy of space than for their excellent performances, as they are less powerful and durable than horizontal engines of the same capacity.

The conical upright frame is bolted to a base of considerable area, which in turn is bolted to the foundation; the cylinder and steam-chest are lagged and banded with brass. The openings to receive the brasses, in the stub ends of the connecting-rod, are cut out of the solid forging. The cross-head guides and pillow-block are cast with the frame, which has the effect of preventing them from be-

* See page 308.

coming loose or getting out of line; and although the cranks are counter-balanced, and the crank-pin, cross-head, wrist, piston, and valve-rod are made of steel, yet the whole design and arrangement of the engine is of a very ordinary and common-place character. The Waters and Crowther's Governor is generally used on these engines.

MASSEY'S ROTARY ENGINE.*

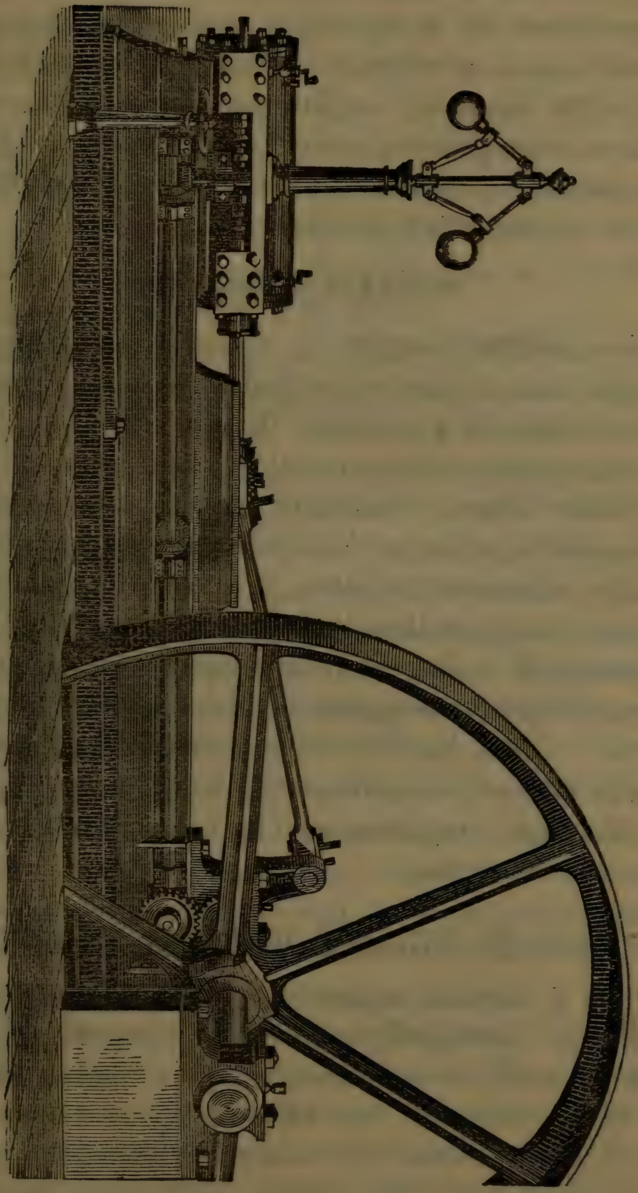
It has been far more frequently the fortune of inventors of rotary engines to fail than to succeed. So frequently, indeed, and from so many various causes, has this been the case, that most engineers adhere to the opinion that the rotary engine can never enter into successful competition with the reciprocating, much less prove a formidable rival. And it is true that until quite recently no rotary engine had been produced that could approach in economy of power the best types of reciprocating engines. This might be attributed to the large amount of clearance, and to the great friction on the journals and packing, and also the leakage caused by wear.

But the difficulties heretofore experienced in the construction and use of the rotary engine—that of keeping them thoroughly steam-tight without undue friction, and that of working steam expansively with a variable cut-off, and without undue clearance—have, it is claimed by the inventor of this engine, been successfully overcome.

The advantages claimed for the rotary over the reciprocating engine are cheapness, lightness, simplicity of parts, compactness, and occupying only a minimum of floor space. The rotary engine, in consequence of its prompt reversing and capability of holding the load, is especially adapted as a hoister for mines and elevators. Besides, it is well suited for working the steering-gear of vessels.

The rotary engine is desirable for locomotive, marine,

* See page 333.



WOODRUFF AND BEACH'S HIGH-PRESSURE CUT-OFF ENGINE.

and stationary purposes, as, when well constructed, one rotary engine can be made to do the work of two reciprocating engines, for in the rotary there are no dead centres or variations in leverage, etc. But still the question, as to what extent the rotary engine can cope with the rotative engine of corresponding power in economical use of steam alone, cannot at present be determined with accuracy, but must be left for future decision.

PORTABLE ENGINES.

By a portable engine is meant a steam-engine so arranged that it may be carried with facility from place to place entire, in a condition for use, either on wheels of its own, or upon a wagon or other conveyance.

It differs from a stationary steam-engine in that the boiler and the engine, with all intermediate and subsidiary parts, are connected together in a compact manner, so as to require no other than their mutual support. While in the stationary engine, the boiler requires a foundation and setting of its own, the engine requires a separate foundation, generally with a detached support for the back end of the main shaft; and not unfrequently the force-pump is apart from the engine, requiring also its independent foundation and source of motion.

HOW TO BALANCE VERTICAL ENGINES.

When a vertical engine runs slow, the weight of the piston and piston-rod, cross-head, connecting-rod, and crank-pin must be counterbalanced, so that it will stand still in any position; but when the speed is very high, it will be only necessary to counterbalance such parts as revolve round the centre of the shaft, the crank-pin, the stub-end, and half the connecting-rod. Very accurate counterbalances must in all cases be determined by trial and experiment.

KNOCKING IN ENGINES.

The causes of knocking in engines are very numerous, and while some of them will yield to an industrious and careful search, others will prove a puzzle alike to the engineer and the expert. Instances are not uncommon where weeks have been devoted, and engines taken all apart and put together again, to find the cause of a knock, when perhaps it was finally discovered to be caused by a loose crank-pin or key in a fly-wheel.

Knocking, in many instances, arises from looseness in the boxes and joints, which strike each other whenever their motion is arrested. Knocking arising from this cause can be easily remedied by taking up the lost motion. In many instances, shoulders become worn in the cylinder in consequence of the piston-rings not overlapping the counter-bore at each end of the stroke. In such cases any adjustment of the piston-packing or keys is generally followed by a knock in the engine. The most practicable remedy for knocking arising from this cause would be to rebore the cylinder.

Knocking is caused in some cases by steam being admitted to the cylinder too late to take up the lost motion when the crank is passing the centre; while in others, in consequence of excessive lead, the steam is admitted too soon and too rapidly, which produces excessive cushioning, and causes the engine to thump.

Engines frequently knock in consequence of the exhaust opening too late and closing too soon. The whereabouts of knocks arising from this cause are generally the most difficult to determine; and it not unfrequently happens that, after all ordinary means have been resorted to in vain, the indicator has to be applied in order to determine the precise location of the knock.

Engines out of line generally knock sideways at certain points of the stroke. The knocking heard in cylinders may be produced by a loose follower-plate or piston-rod; while the noise in steam-chests is generally due to lost motion in the valve connections. Engines in very good condition sometimes knock in consequence of the packing being too hard or too tight around the piston-rod. There are a hundred other causes of knocking which the industrious engineer will be called upon to discover, and while most of them, as before stated, will yield to an easy search, some will try him severely. In fact, to discover knocks, he must see with his *ears* and hear with his *eyes*.

THE INJECTOR.

Of all the inventions of the mechanic and the scientist, nothing seemed to the uneducated to approximate so nearly to perpetual motion as the instrument now in general use as a boiler-feeder on locomotives and stationary engines, and known as the injector, and which, from common use, no longer excites the wonder even of those who do not understand its mode of operation.

It consists of a slender tube, called the steam-tube, through which steam from the boiler passes to another or inner tube, called the receiving-tube. The latter tube conducts a current of water from a pipe into the body of the injector. Opposite the mouth of this second tube, and detached from it, is a third fixed tube, called the delivery-tube. This tube is open at the end facing the water-supply, and leading from the injector to the boiler.

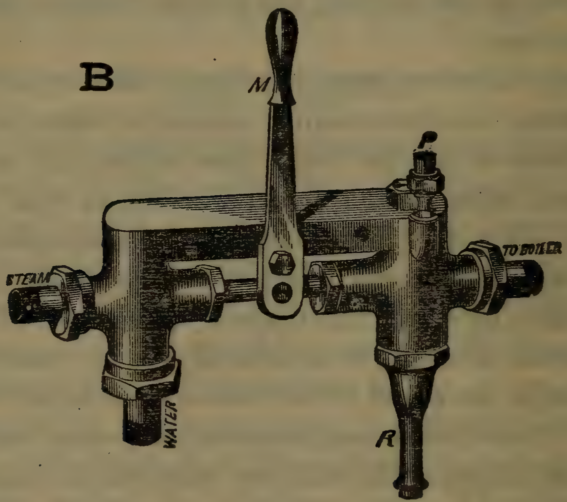
The action of the injector is that which Venturi, in the beginning of the present century, designated as the "lateral action of fluids," and, having been investigated by Dr. Young, in 1805, was proposed by Nicholson, in 1806, for forcing water. The action is identical to that of the

steam-jet, or blower-pipe, in the chimney of the locomotive. The principle is that steam, being admitted to the inner tube of the injector, enters the mouth of a combining tube, in the form of a jet, near the top of the inlet water-pipe. If the level of the water be below the injector, the escaping jet of steam, by its superficial action (or friction) upon the air around it, forms a partial vacuum in the combining tube and inlet-pipe, and the water then rises in virtue of the external pressure of the atmosphere. Once risen to the jet, the water is acted upon by the steam in the same manner as the air had been seized and acted upon in first forming the partial vacuum into which the water rose.

Giffard's discovery was that the motion imparted by a jet of steam to a surrounding column of water was sufficient to force it into the boiler from which the steam was taken, and, indeed, into a boiler working at even a higher pressure. But the most important improvement ever heretofore made in the injector was made in 1868, by Samuel Rue, by which the injector, with steam of from 80 to 90 pounds pressure, is capable of forcing water against a pressure of from 400 to 450 pounds per square inch.

This extraordinary accumulation of power may be explained as follows: The velocity with which steam — say at 60 pounds pressure to the square inch — flows into the atmosphere is about 1700 feet per second. Now suppose that steam is issuing, with the full velocity due to the pressure in the boiler, through a pipe an inch in area, the steam is condensed into water, at the nozzle of the injector, without suffering any change in its velocity. From this cause its bulk will be reduced, say 1000, and, therefore, its area of cross-section — the velocity being constant — will experience a similar reduction. It will then be able to enter the boiler again by an orifice $\frac{1}{1000}$th part of that

by which it escaped. Now it will be seen that the total force expanded by the steam through the pipe, on the area of an inch, in expelling the steam jet, was concentrated upon the area $\frac{1}{1000}$th of an inch, and, therefore, was greatly superior to the opposing pressure exerted upon the diminished area. As the Rue Injector is now successfully employed as a boiler-feeder on the Pennsylvania Line of Steamships, and as it is the only injector that can be used on ocean steamers, river boats, tug-boats, ferries, and pleasure-yachts, a description of the method of its adjustment and working will be of interest to engineers.



Rue's "Little Giant" Injector Letter "B."

How to put on Letter "B" Injector.—Put the injector in a horizontal position above the foot-board, and within easy reach of the engineer, using as short a length of pipe for "steam" and "deliverance to the boiler" as possible. Put an ordinary globe or angle-valve on the steam supply-pipe for starting, etc., taking the steam from the highest part of the boiler, and attaching it to the swivel marked "steam." Attach the water supply-pipe to the

swivel marked "water," putting an ordinary water-cock on the supply-pipe near to the injector. A good supply of water must be had, and if taken from a tank, give it a good fall. The mouth of the pipe should be enlarged, and a screen with small meshes placed over it to keep out dirt; if the supply-pipe be over ten feet in length, or if the water come from a hydrant, or any source that makes a pressure, and the supply is not at a regular pressure, the pipe should be one size larger than the swivel marked "water," which can be done by putting on a reducer. At this point turn on your steam and water, and let them flow through the injector, to see if the pipes and injectors are free from dirt. Then attach the "delivery-pipe" to the swivel marked "to boiler."

Method of Working Letter "B" Injector. — Turn on the water, and, when it flows from the overflow, turn on the steam, slowly at first, until it catches the water, then turn on full head, and push the lever M slowly either forwards or backwards, as seems requisite, until neither steam nor water shows at the overflow. Failure to work will always show at the overflow, and when the point is ascertained at which the lever is to be set for the steam pressure to be carried, it can be regulated, and then left to stand at that position when the steam and water are shut off. The lever is only used to regulate the proportionate amounts of water and steam. But when water is to be lifted by this injector, as on steamships, a small steam-pipe leading from the boiler, and furnished with a valve that opens with a quick motion, is attached to the swivel "P," by means of which a steam-jet is thrown into the tube "R," and the water lifted. But at this point it is necessary to examine the tube in order to ascertain if the suction is good, or if it lifts the water readily, and if so, the steam supply-pipe can be attached to the swivel marked

"steam," and the injector cleared of any dirt that may have collected in the boiler; then the delivery-pipe to the boiler may be attached to the swivel marked "to boiler." Great care should be taken to see that the supply-pipe through which the water is lifted is perfectly air-tight, as any leak in the pipe will interfere with the working of the injector.

In ordering injectors, it should always be stated whether the connecting-pipes are copper, brass, or iron, and whether for steamships or stationary boilers.

TABLE

OF CAPACITIES OF RUE'S "LITTLE GIANT" INJECTOR.

Size of Injectors.	Size of Pipe Connections.	Pressure of Steam in Pounds.	Gallons. Per Hour.	Nominal Horse-Power.
0	$\frac{1}{4}$	90	60	4 to 8
1	$\frac{3}{8}$	90	90	6 " 12
2	$\frac{1}{2}$	90	120	8 " 20
3	$\frac{3}{4}$	90	300	20 " 40
4	1	90	600	40 " 80
5	$1\frac{1}{4}$	90	900	60 " 120
6	$1\frac{1}{4}$	90	1200	80 " 160
7	$1\frac{1}{2}$	90	1620	140 " 225
8	2	90	2040	200 " 275
9	2	90	2480	250 " 350
10	2	90	3000	300 " 400
12	$2\frac{1}{2}$	90	3600	350 " 500

PUMPS.

Pumps may be divided into two classes, — "lift" and "force," although some pumps perform the double duty of lifting and forcing. "Lift" pumps cause the water to raise itself by having its surface relieved of the column of air resting upon it.

The surface of all water exposed to the air has the

pressure of the air or atmosphere resting upon it; if, therefore, one end of a pipe or tube be lowered into water, and the other end be closed by means of a valve or other device, and the air contained in the pipe be drawn out, it is evident that the surface of the water within the pipe will be relieved of the pressure of the atmosphere; and there will be no resistance offered to the water to prevent its ascending the pipe. The water outside of the pipe, still having the pressure of the atmosphere upon its surface, therefore forces water up into the pipe, supplying the place of the excluded air. The water inside the pipe will rise above the level of that outside of the same in exact proportion to the amount to which it is relieved of the pressure of the air, so that, if the first stroke of a pump reduce the pressure of the air contained in the pipe from 15 pounds on the square inch (which is its normal pressure) to 14 pounds per inch, the water will be forced up the pipe to the distance of about 2¼ feet, because a column of water an inch square and 2¼ feet high is equal to about 1 pound in weight.

It is evident that, upon the reduction of the pressure of the air contained in the pipe, from 15 to 14 pounds per square inch, there would be (unless the water ascended the pipe) an unequal pressure upon its surface inside as compared to that outside of the pipe; but in consequence of the water rising 2¼ feet in the pipe, the pressure on the surface of the water, both inside and outside, is evenly balanced (taking the level of the outside water to be the natural level of the water inside), for the pressure upon the water exposed to the full atmosphere will be 15 pounds upon each square inch of its surface; while that upon the same plane, but within the pipe, will sustain a column of water 2¼ feet high (weighing 1 pound) and 14 pounds pressure of air, making a total of 15 pounds, which is,

therefore, an equilibrium of pressure over the whole surface of the water at its natural level.

If, in consequence of a second stroke of the pump, the air pressure in the pipe is reduced to 13 pounds per inch, the water will rise up it another $2\frac{1}{4}$ feet, and so on until such time as the rise of a column of water within the pipe is sufficient to be equal in weight to the pressure of the air upon the surface of the water without; hence it is only necessary to determine the height of a column of water that will weigh 15 pounds per square inch of area at the base of the column to ascertain how far a suction-pump will cause water to rise, and this is found by calculation or measurement to be a column nearly 34 feet high. But even this height is partly theoretical, as it may be said that the extreme practical limit to the elevation of water by lift- or suction-pumps, at the ordinary level of the sea, is 30 feet; but more satisfactory and sure results will be obtained if the pump is not made to lift over 20 to 25 feet, and even then extreme care must be taken that the suction-pipe is absolutely air-tight, and the packing in the pump tight and in good condition.

Force-pumps are those by means of which the water is expelled from the pump-barrel and through the delivery-pipe by means of the mechanical force applied to the pump-piston or plunger; the amount of power required to drive such a pump will, therefore, depend at all times upon the height to which the water is required to be forced. When a pump is arranged to draw the water, and force it after it has left the pump-barrel, it is termed a lift- and force-pump; but if the water merely flows into it in consequence of the level of the water supply being equal to or above that of the top of the pump-barrel, it is termed simply a force-pump. Hence a suction-pump performs its duty in causing the water to rise to the pump; a

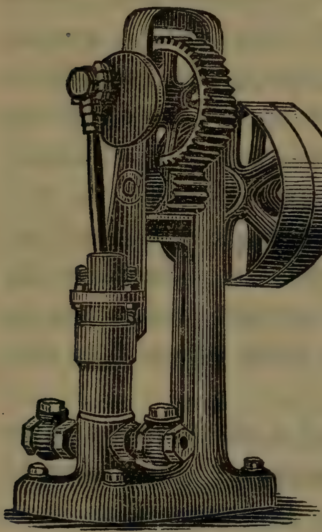
force-pump is one which performs its duty in expelling water from its barrel, and a suction- and force-pump is one which performs both duties alternately.

No pump will lift very hot water, for the reason that, in attempting to do so, the atmospheric pressure being taken from it, it passes into vapor and steam, which passing through the suction-pipe into the pump, fills the cylinder with vapor instead of water, so that on the return stroke, the piston meeting with no resistance, moves rapidly until suddenly striking the water, which partially fills the cylinder, a violent concussion is produced, which is very injurious to the pump and its connections. Therefore, for pumping hot liquid the point of supply should not be below the pump, but if possible a little above it, so that the liquid may flow into it.

Piston-pumps.—The capacity of a piston-pump is its area multiplied by the length of its stroke; but it must be remembered that all pumps throw less water than their capacity, the deficiency ranging from 20 to 40 per cent., according to the quality of the pump. This loss arises from the lift and fall of the valves, from inaccuracy of fit or leakage, and in many cases from there being too much space between the valves and piston, or plunger. The higher the valves any pumps have to lift, to give the necessary opening, the less efficient the pump will be.

An air-chamber placed in the suction-pipe of any pump causes a better supply of water to the pump by holding a body of water near to it, and by making the supply of water up the suction-pipe more uniform and continuous. Air-chambers should be made as long in the neck as convenient, so that the water in passing through the pump-barrel to the delivery-pipe could not be forced up into the chamber, as, if such should be the case, the air in the chamber is soon absorbed by the water, and consequently

the supply of water diminished. If an air-chamber be placed on top of a pump-barrel or in the delivery-pipe, it has a tendency to facilitate the delivery, and also to cushion the piston and modify the jar that would otherwise result from the piston striking against a solid body of water.



The annexed cut represents an ordinary lift- and force-pump, with trunk-plunger and two valves, the pipe from one leading to the well or cistern, and from the other to the boiler or tank.

In the ascending stroke of the plunger the suction-valve is opened; and if the air be expelled from the barrel of the pump, and a perfect vacuum be formed in the pump-barrel and pipe, the water will follow the piston about 33 feet. In the descending stroke, the lift-valve is closed by the compression of the water in the barrel of the pump, caused by the downward movement of the plunger, which, at the same time, causes the discharge-valve to open, and forces the water out into the boiler or tank.

Boiler feed-pumps should, in all cases, have a capacity equal to one cubic foot of water per horse-power per hour, and as water expands by heating, a pump that would be large enough to furnish a given quantity of cold water, would not be of sufficient capacity if the water was heated.

Pumps located near boilers are very liable to become hot; and in such cases, if the pump should fail to lift, the pet-cock in the pump-cylinder should be opened, in order to allow the hot water in the pump-barrel to run out and lessen the pressure, when it will be found, in most cases, that the pump will work.

It frequently happens that when pumps, to all appearances, are performing well, the water does not rise in the boiler or tank, but gradually falls below the level at which it stood when the pump was started, which may be due to any of the following causes: leakage in the suction-pipe, wear or looseness of the packing, the water- and check-valves being kept from their seats by shavings, straw, or other foreign substances, drawn in through the suction-pipe; the water supply being cut off, or the pipes becoming choked with lime, salt, and such mineral substances as are commonly found in spring, river, or sea water.

In order to arrive at some definite conclusion as to the direct cause of the difficulty, the check-valve should be first examined to see if it is in operation; this can be proved by applying the ear to the valve, and ascertaining if it rises and falls at each stroke of the pump: the action of the valve can also be determined by applying the hand to the feed-pipe below the valve; or the stop-cock between the check-valve and the boiler may be closed, and the check-valve taken out; then by allowing the pump to make a few strokes, any sediment that may have found its way into the pump-barrel or under the check-valve will be washed out. The ends of the suction-pipes of all pumps that draw water from wells, creeks, rivers, or tanks should be covered with a screen or strainer.

All pumps, whether hot or cold water, bilge, independent, or auxiliary, should be frequently examined and tried by being put in service, in order that they may be ready for any emergency that may arise.

Pumps should be kept perfectly clean and free from oil and dirt, as such accumulations detract very much from their general appearance, and interfere very materially with their working.

STEAM-PUMPS.

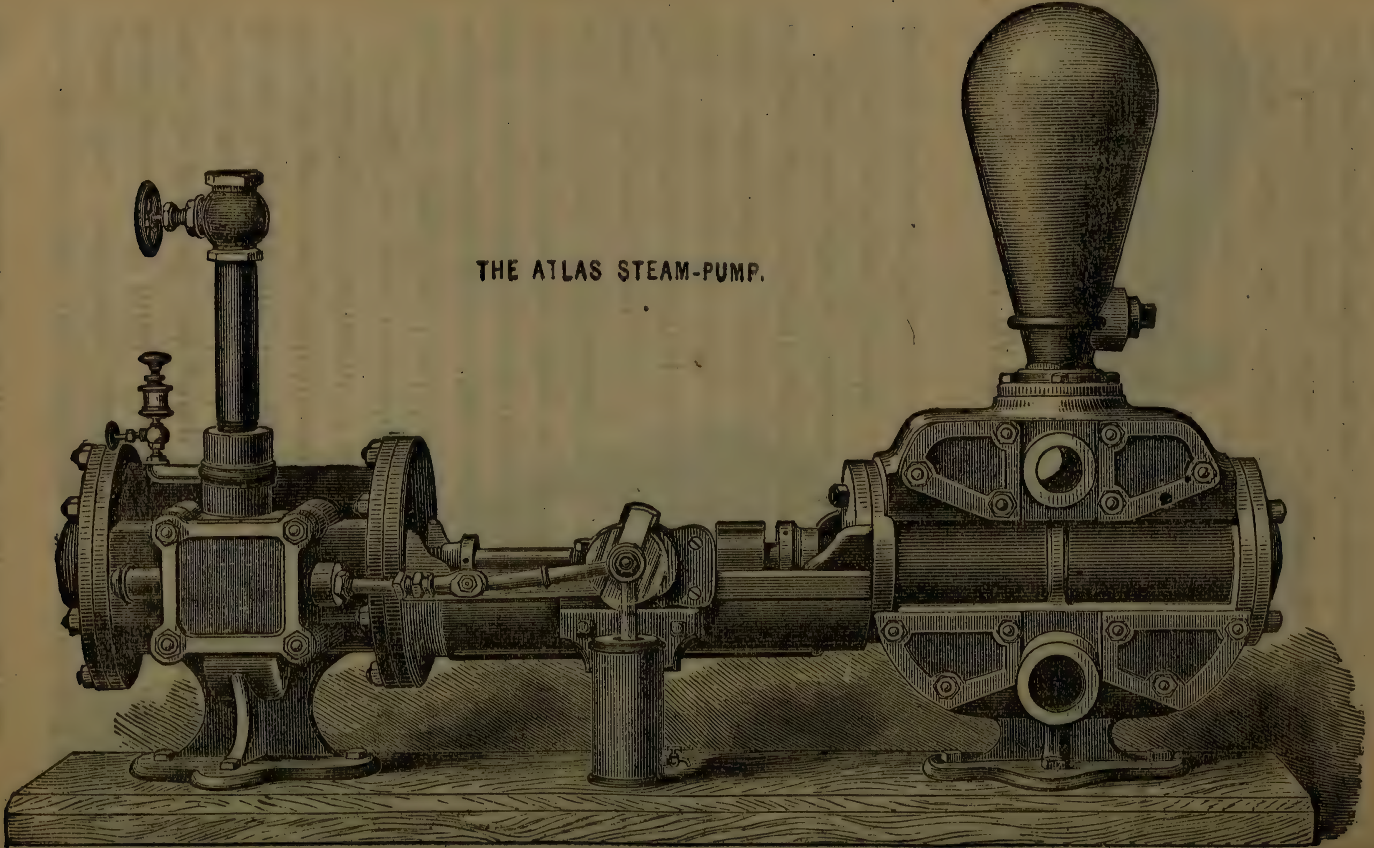
Steam-pumps may be said to be among the most essential requisites of the age. They are used for boiler-feeders, extinguishing fires, pumping liquids, and, in short, for almost every variety of purpose in manufacturing and in civil and mechanical engineering. As boiler-feeders, they are superior to any other known invention, as they are available for pumping feed-water against extremely high-pressures, and for keeping boilers supplied when circumstances require the stoppage of the engine. They can be regulated either to furnish a constant supply of water for boilers or other purposes, or the supply can be largely increased if it should be found necessary to meet some unforeseen emergency.

The cut on page 179 represents the Atlas steam-pump, which is especially adapted for fire, marine, mining, tanning, and wrecking purposes; or, in fact, for any place where it is desirable to displace large quantities of water in a short time, such as filling tanks and reservoirs, draining mines or quarries, or freeing the holds of vessels from water in case of leakage, etc.

It is a double direct-acting pump; in its construction the common slide-valve is used, and operated by a peculiar rocker and cam motion, and is so constructed that it is impossible for it to stop on a dead-centre, for as soon as the motive power is admitted to the cylinder (let it be steam, water, or air pressure), the pump begins to operate, and it is impossible to place the valve in such a position as to shut off steam and stop the pump.

By a peculiar arrangement for moving the steam-valve, a full stroke is insured, but at the same time, by means of a guide to the valve motion, the stroke is slowed down, thus giving the water-cylinder time to fill, insuring a full

THE ATLAS STEAM-PUMP.



stream every time, and preventing the plunger or water-piston from cushioning against the water when the cylinder is but partly filled. The water-valves are made especially for the work the pump is required to do, generally of gun-metal, or of such other material as is least affected by the liquid passing through them. The openings are straight and clean, without small or intricate passages to be filled up with sediment or dirt.

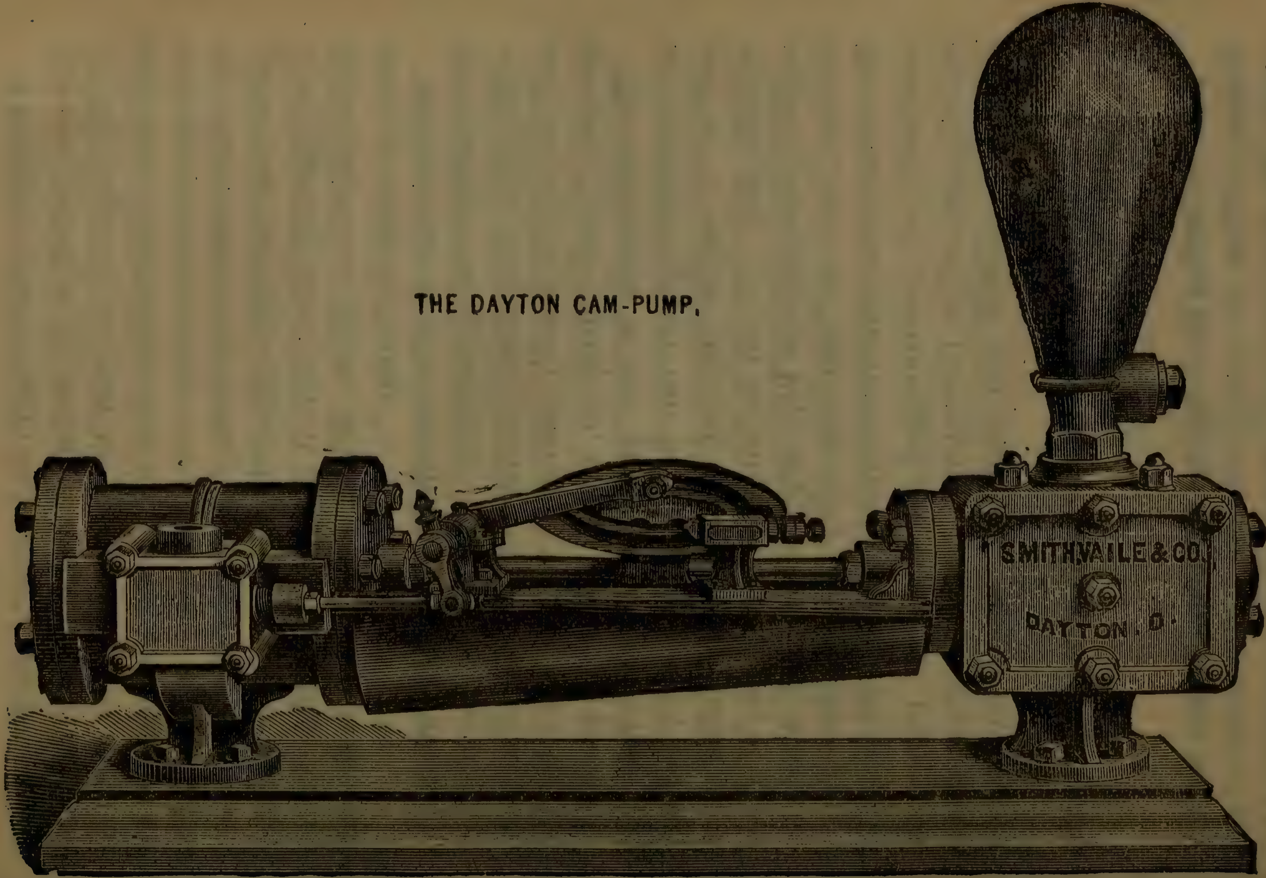
Among the most valuable features of this pump are its great simplicity, durability, and effectiveness, as it is equally reliable whether used every day or once in a year; and even when submerged in a mine or the hold of a ship, it will start when the steam is turned on from the boiler, and will work notwithstanding the condensation of the steam incident to carrying it under water. Its parts are few, and are so constructed as to show hardly any perceptible wear even after being used for years. No well-regulated manufactory, steam-using establishment, or steam-ship, should be without one or more of these pumps.

There are several thousands of them in use in the United States and Canada. They are made of any size, and to discharge from 1 to 1000 gallons per minute.

THE DAYTON CAM-PUMP.

On page 181 is shown a cut of the Dayton Cam-Pump. The inventor of this pump seems to have studied all the requirements of a good steam-pump, and to have avoided all points of defect from a liability to wear, or otherwise, in its construction, thus rendering it one of the most simple, durable, and reliable pumps in the country. It has been fully tested, within the past three years, under the most varying conditions, and its superiority fully established.

THE DAYTON CAM-PUMP.



The steam-valve is a plain slide-valve, worked by means of a cam bolted on the piston-rod and moving with it, and by the shape of the cam the stroke is slowed down at each end, giving ample time for the water-cylinder to fill and water-valves to close before the return stroke, insuring a full stream every stroke, and preventing the pump-piston from striking against the water when the cylinder is but partially filled; and it is utterly impossible for the steam-valve to be thrown in such a position as to shut off steam and stop the pump.

Another feature of great importance is, that this pump has absolutely no dead-centre; whatever be the position of the parts, when steam is turned on, it is bound to start, and this with a velocity absolutely in proportion to the amount of steam turned on, and, of course, its pressure.

The mechanism of this pump is also so simple that any ordinary engineer can fully comprehend it, and any part of it can be repaired without being taken to a machine-shop. It can be run at a higher rate of speed without thumping, and kept in perfect running order with less cost, than any plain side-valve pump in the country. The obstacles and annoyances arising from the complicated machinery and uncertainty of action in other pumps are entirely obviated in this pump. Hence its great popularity with railroad men, engineers, miners, and manufacturers generally.

The Atlas and Cam Pumps are manufactured by Smith, Vaile & Co., Dayton, Ohio.

Rule for finding the Diameter of Pump-plunger for any Engine.—When the pump-stroke is ½ the stroke of the engine, the diameter of the steam-cylinder multiplied by 0·3 will give the proper diameter of pump-plunger.

Another Rule.—When the pump-stroke is ¼ the stroke

of the engine, the diameter of the cylinder multiplied by ·42 will give the proper diameter of pump-plunger.

Diameter of pump-plunger should be equal to $\frac{1}{3}$ the diameter of the cylinder; when the pump-stroke is $\frac{1}{2}$ the engine-stroke.

Diameter of pump-plunger should be equal to $\frac{1}{6}$ of the diameter of the cylinder, when the pump-stroke is $\frac{1}{4}$ the engine-stroke. The velocity of water in pump passages should not exceed 500 feet per minute. Pump-valves should have an area of $\frac{1}{4}$ the area of the pump.

Feed-pumps for Condensing Engines.—For condensing engines, the diameter of the pump-plunger should equal 1·11 the diameter of the steam-cylinder when the pump-stroke is $\frac{1}{2}$ the engine-stroke, and $\frac{1}{8}$ the diameter of steam-cylinder when the pump-stroke is $\frac{1}{4}$ the stroke of the engine.

Rule for finding the Necessary Quantity of Water per Minute for any Engine.—Multiply the cubic space in cylinder in inches, to which steam is admitted before being cut off, by twice the number of revolutions per minute, and divide the product by the comparative bulk of steam * at the pressure used; the quotient will be the cubic inches of water required per minute.

EXAMPLE.

Diameter of cylinder, 12 inches. Area....................	113·09 sq. in.
Stroke, 24 in. Steam cut-off at $\frac{1}{2}$ stroke....................	12 in.
Revolutions per minute..	60
Pressure per sq. in., 70 lbs. Cubic in. steam from 1 cubic in. water..	408

$$
\begin{array}{r}
113{\cdot}09 \\
\underline{12} \\
1357{\cdot}08 \\
\underline{120} \\
408)\overline{162849{\cdot}60} \\
399{\cdot}14 \text{ cubic inches of water.}
\end{array}
$$

* See Table on pages 39–43.

This rule takes into account the expenditure of steam only; but, as is well known in practice, a large quantity of water passes from the boiler to the cylinder in mechanical combination with the steam, allowance must therefore be made for such losses, also for the waste incurred by clearance in the cylinder, cubic contents of steam-ports, condensation, etc., so that in the selection of a pump for any engine, it is advisable that it should be of sufficient capacity to furnish at least twice the quantity of water designated by the rule.

The capacity of any pump can easily be determined, if its dimensions are known, by the following rule: Multiply the area of piston, in inches, by its stroke in inches, and this product will give the capacity in cubic inches per single stroke; divide this by 231, and the result will be gallons per single stroke.

The power required to raise a given quantity of water a certain height can be easily computed by the following rule: Multiply the amount of water, in gallons, to be raised per minute by 8·35 (the weight of a gallon of water), and this product by the height, in feet, of the discharge from the point of suction; divide the result by 33,000, which will give the theoretical horse-power required to raise the amount of water a certain distance. But from this result there should be an allowance made of from 10 to 30 per cent., for loss induced from leakage in the pipes, short-bends, bad condition of the pump, friction of water in the pipes, friction of the parts of the pump in contact, etc.

The speed at which pumps should run must depend entirely upon the purpose for which they are to be used. The speed of boiler-feed pumps should range from 20 to 30 double strokes per minute, according as the water is hot or cold; but they should never exceed 40 strokes.

DIRECTIONS FOR SETTING UP STEAM-PUMPS.

Pipes fully as large as the pump connections should be used in all cases, and where it becomes necessary to use long or crooked pipes, they should be even larger.

The discharge-pipe should be the same diameter from the pump to the boiler, as any reduction in the diameter of a pipe greatly diminishes its capacity.

A pipe two inches in diameter, 100 feet long, will deliver but $\frac{1}{4}$ the quantity of water that a pipe 2 inches in diameter and 2 inches long will with the same pressure.

Short bends and angles in the pipes should be avoided as much as possible, as they retard the flow of the water; but when they have, of necessity, to be used, they should be as large as practicable.

Leaks in the suction-pipes of pumps should be positively guarded against, as a very small leak will destroy the efficiency of a good pump.

The exhaust-pipe of steam-pumps should be run down, when convenient, in order that the water of condensation may flow out instead of being forced out.

The pipes of all pumps located in exposed situations should be furnished with unions, in order that they may be separated from the pumps on extremely cold nights; and the drip-cocks in the steam- and water-cylinders should invariably be left open at night. All pumps require more care in winter than in summer.

THE PULSOMETER.

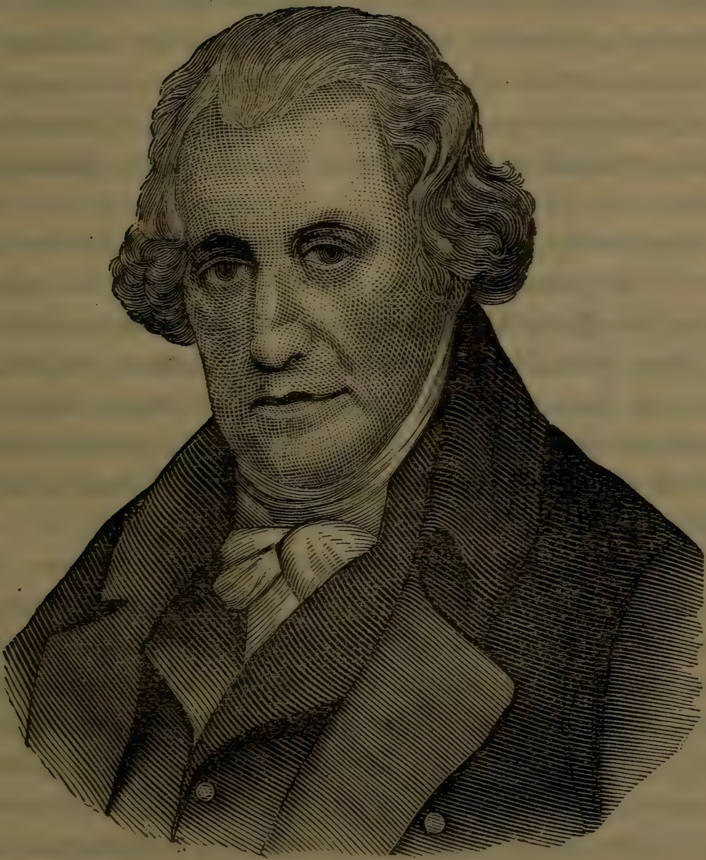
The conditions upon which the proper action of the pulsometer depends are similar, in all essential particulars, to those which pertain to the management of the ordinary double-acting piston pump; but although the pulsometer

may be operated under the same conditions as control the operation of piston-pumps; yet the arrangements should be made with judgment and discretion, to meet the characteristic difference existing between the two systems, and with special reference to the nature of the fluid to be pumped.

The pulsometer requires a free current of the fluid in an uninterrupted stream, as a necessary condition of its action. If the source of supply of the liquid be exhausted, or the induction passage is contracted so as to prevent its free admission in response to the vacuum impulses alternately created in the chambers above, the steam will blow through the discharge-pipe, and the pulsation will cease.

The pulsometer is peculiarly adapted for pumping water from mines, foundations, or excavations where quicksand or mud occur, as it will pump water combined with fifty per cent. of mud or sand without any derangement of its parts, as there is no cylinder, piston, or valves to cut or wear. It can also be used to irrigate land, drain swamps and ponds. It is also available as a bilge-pump on board of vessels, as it is not liable to become choked with grain or other substances. The pulsometer will raise water or any other liquid from a depth due the vacuum produced by the condensation of steam, and will force the same to an elevation due the initial pressure of the steam in the boiler operating it.

The pulsometer possesses some advantages in point of economy and convenience, as it can be lowered into deep wells or mining-shafts, and, in fact, set or hung up in any place most convenient to the work or the steam; but it is less reliable in its action than either the steam-pump or the injector, and is far more wasteful of steam.



JAMES WATT.

A name which must endure while the peaceful arts flourish.

CONDENSING, OR LOW-PRESSURE STEAM-ENGINES.

When a steam-engine is so arranged that the exhaust steam from the cylinder escapes into a condenser and is condensed into water, it is termed a condensing, or low-pressure engine; because, by condensing the steam and producing a vacuum, not only is the pressure of steam in the boiler, but also the 15 pounds per square inch required in the non-condensing engine to overcome the pressure of the air, rendered available as an effective force against the piston.

Consequently, when a condensing engine is working steam of 10 pounds pressure per square inch by the steam-gauge, the effective pressure per square inch exerted on the piston would be 25 pounds if the vacuum is perfect, which in fact, it never is; the average being about 12 pounds, which, with the boiler pressure of 10 pounds, would exert an effective pressure on the piston of from 21 to 22 pounds per square inch.

In Watt's engines the ordinary working pressure was from 5 to 6 pounds above that of the atmosphere; but the steam-valve remained open and allowed steam of the boiler pressure to follow the piston during the whole length of the stroke. Now, however, there are few condensing engines run with less than from 20 to 25 pounds per square inch boiler pressure; but the steam is generally cut off from the cylinder at one-half the stroke, and then allowed to expand during the remainder.

It may be fairly assumed that a non-condensing engine has, on an average, at least two pounds per square inch back pressure on the piston. Some have much more than this, and first-class engines have less; but two pounds can be considered a fair example of ordinary practice. By the application of the condensing principle, there would

be a negative pressure of say 10 pounds per square inch on the back of the piston, so that the piston pressure would be increased by 12 pounds.

Now, if the application of the condensing principle decreases the back pressure, it is obvious that it must increase the positive pressure on the piston. It is plain, therefore, that a lower pressure of steam can be used, or, that the steam may be cut off at an earlier point of the stroke; the gain in either case can be approximately calculated. If the gain in positive pressure produced by the reduction in back pressure be multiplied by 100, and divided by the mean effective pressure, it will give the percentage of gain due to the effect of the condensing principle.

It is tolerably safe to estimate the saving effected by the use of the condensing over the non-condensing engine at from 20 to 25 per cent. of the amount of steam used, and consequently the amount of coal consumed. It sometimes happens, however, that little or no saving is effected in the condensing engine; but this loss is not in the principle, but is in nearly all cases induced by defects in the machinery, which admit of a back pressure sufficient to counterbalance the effect due to the vacuum.

To get the maximum of economy out of any class of expansive condensing engine, the pressure of steam and point of cut-off must be so regulated that the steam passes into the condenser at the end of the stroke at a pressure not exceeding 5 pounds above a perfect vacuum; and with steam at 45 pounds pressure above the atmosphere, which is equal to 60 pounds pressure above a perfect vacuum (the pressure of the atmosphere being considered as equal to 15 pounds on the square inch), and a terminal pressure of 5 pounds, we get 12 expansions, because the pressure at the end of the stroke is 12 times less than what it was at the points of cut-off.

In condensing engines, the expenditure of steam—and therefore the performance — is generally estimated by what is termed the weight of sensible steam in the cylinders; that is, the weight as shown by the indicator diagrams. This indication, in engines in which the condensation in the cylinders is limited by suitable conditions of action, may approach the real figure, so as to give results accurate enough in practice; still, in other cases, it gives results of no possible value.

This weight of sensible steam in the cylinder differs from the weight of water actually vaporized in the boiler, because it does not take into account the loss of steam by leakages, condensations in the pipes, or the dead spaces at the cylinder's ends, etc. Then, again, it is no uncommon thing to see the consumption of fuel indicated with the minutest accuracy in figures containing two or three decimals, and this without the slightest mention being made of the kind of fuel used; though it is well known that the heating power of some kinds of coal is nearly, if not fully, twice that of others.

EXPLANATION OF THE WORKING PRINCIPLES OF THE CONDENSING ENGINE.

The steam-valve being opened, steam flows through the passage to the cylinder and forces the piston down or forward, as the case may be; while, at the same instant, the exhaust-valve opens, and allows the steam that is in front of the piston, to escape through the exhaust-pipe into the condenser, where it mingles with the injection water* and is condensed into water, thereby forming a vacuum in front of the piston; consequently, the whole pressure on

* The steam mingles with the injection water in the jet-condenser only as it is condensed by being brought in contact with cold surfaces, when the surface-condenser is used.

the opposite side of the piston is effective pressure. When the piston reaches the end of the stroke, the steam-valve closes, and the opposite steam- and exhaust-valves open, the steam forcing the piston forward, as in the former case, and the exhaust steam escaping to the condenser.

The condensed water which collects in the bottom of the condenser is drawn off through the foot-valve to the air-pump, and forced out through the delivery-valve into the hot-well, from which it is taken to feed the boilers or delivered overboard. Thus, it will be seen that the circulation from the boiler through the cylinder and condenser to the hot-well, and back again to the boiler, must be continually kept up while the engine is in motion.

A condenser and air-pump can be attached to any non-condensing engine, and it then becomes what is commonly known as a low-pressure engine. In this case the initial pressure of the steam can be reduced, or the valve may be altered so as to cut off the steam at an earlier point of the stroke; and the engine will still develop the same power as before.

HORSE-POWER OF CONDENSING ENGINES.

In Great Britain, every steam-vessel is rated at a certain nominal horse-power; in the United States, on the contrary, the dimensions of the cylinder are almost universally expressed in feet and inches, the nominal power being considered of little moment, and the actual power being well understood to depend on the effective pressure in the cylinder, and the speed of the piston, both of which elements may be varied at will within certain limits.

Watt's Rule for calculating the power of condensing engines is as follows:—"Multiply the square of the diameter of the cylinder in inches by the cube root of the

stroke in feet, and divide the product by 47; the quotient is the number of nominal horse-power." This rule supposes a uniform effective pressure upon the piston of 7 pounds per square inch, and a piston-speed of 160 feet per minute.

The following Rule is more applicable, however: 1st. Find the mean effective pressure in pounds upon the piston. 2d. Find the pressure upon the piston, in pounds, necessary to overcome the friction of the engine, also that necessary to overcome the friction of the air-pump: deduct these two amounts from the whole mean effective pressure; then multiply the remainder by the velocity of the piston in feet per minute. 3d. Find the atmospheric pressure in pounds upon the air-pump bucket and multiply it by the velocity of the air-pump bucket in feet per minute; subtract this amount from the first product and divide the remainder by 33,000; the quotient will be the actual horse-power of the engine.

The losses arising from the different causes are shown by the following table, the pressure of steam in the boiler being represented by 1.000.

Loss from velocity of steam into cylinder	·007
Loss through condensation of steam in cylinder and pipes	·016
Loss occasioned by passing through openings and pipes	·007
Loss occasioned by cutting off steam	·100
Friction of piston, and loss by waste through packing	·125
Force required to move the different valves, and friction of bearings	·063
Power required to work the air-pumps	·050
Total loss	·368
Remainder	·632
Counter-pressure occasioned by vacuum	·063
Multiplier in full for resistance	·695

This multiplier may be used in calculating the actual

power of all condensing engines from 100 to 2000 horse-power.

EXAMPLE.

Diameter of cylinder, 70 inches = Area, 3848·46 square inches.

$\left.\begin{array}{l}\text{Stroke, 14 feet,}\\ \text{No. of revolutions, 16,}\end{array}\right\}$ = Travel of piston per minute, 448 feet.

Pressure, 33 pounds per square inch. Cut off at 9 feet.

14 ÷ 9 = 1·5 Hyperbolic Logarithm of 1·5 = 40546.

$·40546 + 1 = 1·40546.\quad \dfrac{1·40546 \times 33}{1·5} = 30·92$ mean pressure per square inch.

$$\begin{array}{r} 30·92 \\ 3848·46 \\ \hline 118994·3832 \\ ·448 \\ \hline 53409483·6736 \\ ·695 \\ \hline 33{,}000)37119591·1531520 \\ \hline 1124·83 \text{ horse-power.} \end{array}$$

THE VACUUM.

The literal meaning of the term "vacuum" is space unoccupied by matter. The cylinder of a steam-engine filled with steam, though vaporized from a small quantity of water, cannot be said to be void of matter; but condense that steam to its original bulk into water, and withdraw this water from the cylinder, and the space formerly occupied by the steam will be unoccupied. No matter remaining in the cylinder, there is what is termed a "vacuum," or a void space.

We have supposed this operation to have taken place under the piston of a steam-engine, and in that case there is no resistance to be overcome in the descent of the piston.

The pressure of the atmosphere alone, which is 15 pounds to the square inch, or thereabouts, would suffice to force the piston down with a power equal to the degree of

vacuum formed up to the limit stated,—15 pounds, if the vacuum be perfect.

The first steam- and atmospheric-engine was constructed on this principle: Steam being admitted below the piston in a cylinder with its upper end open to the atmosphere, which caused it to rise; the steam being then condensed in the same cylinder by the application of cold water; the piston being forced down on the return-stroke by the pressure of the atmosphere alone. If steam be allowed to take the place of the atmosphere, as in Watt's engine, steam at atmospheric pressure will produce the same effect. 1700 cubic inches of steam, or one cubic inch of water converted into steam at atmospheric pressure, will have a force sufficient to raise a weight of one ton one foot high; two cubic inches of water, two tons; and each additional ton, one cubic inch of water converted into steam.

One of Watt's first improvements was to attach to the steam-engine a second vessel, in which to condense the steam. This he called a "condenser." He also enclosed the upper end of the cylinder with a cover, by which arrangement steam was admitted to each side of the piston, and made to answer a double purpose. And by connecting each side of the cylinder with the condenser, the condenser being supplied with cold water, the vacuum was very much improved; thus the power of steam at a pressure above the atmosphere was made available, in addition to the pressure of steam below the atmosphere. By admitting the steam alternately on each side of the piston, after a partial vacuum had been formed in the condenser, the engine was thereby made to be double-acting, and its power increased in proportion to the pressure of the steam above the pressure of the atmosphere.

Steam arising from an open vessel, for instance, from

the man-hole of a steam-boiler, has a force greater than the pressure of the atmosphere, inasmuch as it has to displace the atmosphere before it can rise above the surface of the water. The resistance of the atmosphere is equal to about 15 pounds on the square inch. It varies from $13\frac{1}{2}$ pounds to upwards of 15 pounds; on the average, it is about $14\frac{3}{4}$ pounds to the square inch. 2·037 inches of a column of mercury balance 1 pound pressure to the square inch. Therefore, steam enclosed in a steam-boiler, at 5 pounds pressure per square inch above the atmosphere, or, in other words, at 5 pounds on the steam-gauge, is in reality a pressure of 20 pounds on the square inch, as applied to the piston of a steam-engine under the conditions above stated—taking the pressure of the atmosphere at 15 pounds. If the pressure be less, as it often is, say 14 pounds, then the pressure upon the piston would be 19 pounds, because the resistance of the atmosphere on the safety-valve and steam-gauge would be less, and the steam in the boiler also less, in proportion to the reduced pressure of the atmosphere.

Hence it arises that an engine heavily loaded varies in its speed with the varying pressure of the atmosphere. Suppose that the vacuum is not perfect,—and in practice it never is so,—and that there remains in the cylinder a portion of uncondensed steam, the resistance of which is equal to 3 pounds to the square inch, then the steam on the upper side of the piston, at 5 pounds to the square inch above the pressure of the atmosphere, would act with an effective force of 17 pounds upon the square inch: the upper side of the piston having exerted upon it a pressure equal to 20 pounds to the square inch, and the under side a pressure or resistance equal to 3 pounds to the square inch.

Under these circumstances, the condenser will have ex-

hausted steam from the cylinder equal to 12 pounds to the square inch, commonly termed a 12-pound vacuum; and the uncondensed steam which has been left in the cylinder will have a resisting force equal to 3 pounds to the square inch. In proportion to the quantity of steam condensed to the whole, so is the value or available pressure upon the piston. If the uncondensed steam left in the cylinder was equal to 6 pounds to the square inch, then, in the other circumstances supposed, the available pressure upon the piston would be only 14 pounds to the square inch — a proof that vacuum is not power, as many are led to suppose.

All power in the steam-engine is derived from the pressure of the steam upon the piston. If there be no resistance on the other side of the piston, the whole pressure is available; when there is resistance, whatever be the amount, it has to be deducted. The available power of steam on the piston is what is left of the whole force when that deduction is made. All power from air, steam, or gas is the result of pressure or density; and in proportion to the pressure so is the power.

If 5 pounds more pressure to the square inch be added to the 5 pounds before described, the pressure of the steam above the atmosphere will be 10 pounds to the square inch, making the available pressure 25 pounds. Many suppose that by doubling the pressure above the atmosphere, or double the pressure as shown by the steam-gauge, that the power of the engine is doubled. Such, however, is not the case; for, though the pressure of the steam in the boiler above the atmosphere is doubled, they have only added 5 pounds to the 20 pounds already available. This only gives 25 pounds to the square inch upon the piston of the engine, in place of 20 pounds. The increased power of the engine is as 25 is to 20, supposing

the non-resistance, or, in other words, the vacuum, to be the same. In the practical working of engines, it is seldom that the full pressure in the boiler can be brought to bear upon the piston. To ascertain the real pressure operating on the piston at any given time, the *indicator* has to be resorted to, in the manner hereinafter explained. The foregoing conclusions have reference entirely to the CONDENSING ENGINE.

Should the vacuum become impaired, which is often the case, the first thing to be done is to screw down the glands of the stuffing-boxes of the piston, air-pump, and expansion joints; then hold a lighted candle near them, and observe whether the flame is drawn in or not. If all the joints are found to be tight, it is evident that the diminution in the vacuum is due to some other cause; perhaps an insufficiency of injection water, or some derangement of the foot- or delivery-valves.

MARINE STEAM-ENGINES.

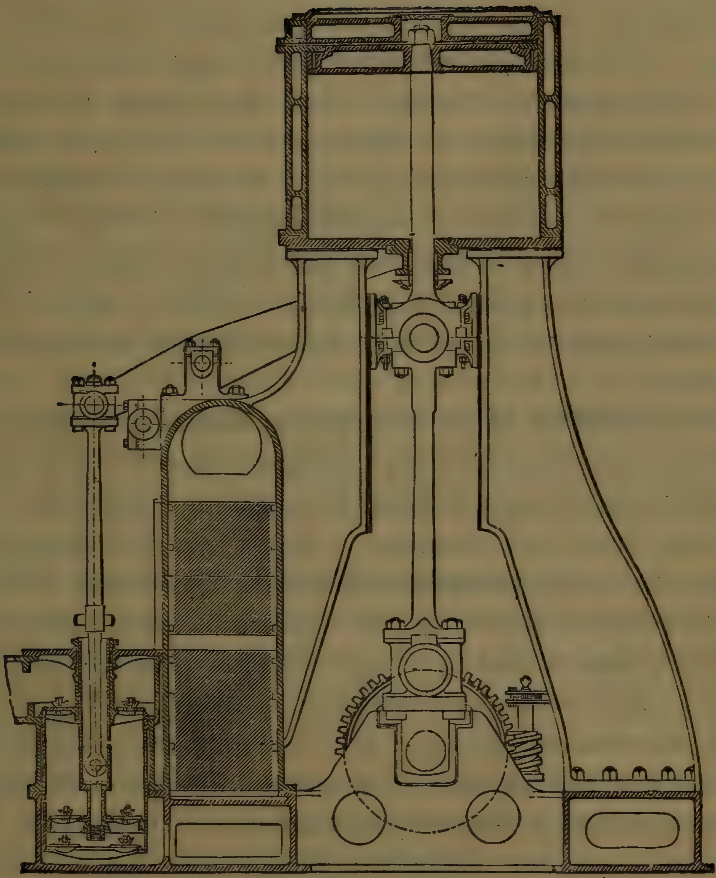
The marine engine is one the construction of which is modified so as to enable it to be placed in a vessel, and, by working, to propel it through the water. In land engines the machinery is not, in general, limited in the space it occupies; part of it may, if more convenient, be placed underground. This frequently happens with the condenser, air-pump, and cylinder. But it is quite different in the case of the marine engine, as the sleepers in the bottom of the vessel determine the depth at which the engine can be located, and the upper deck, in most cases, limits the height; moreover, the less space occupied by the engine, the more can be devoted to the accommodation of the crew, passengers, and cargo.

In considering the various engines proposed or in use

for marine purposes, either as respects form or detail of construction, the questions that most interest us refer to space occupied, accessibility of parts for cleaning, oiling, or repairing, loss of heat induced by radiation, degree of friction, tendency to jerk or jar, liability to be thrown out of line, free admission and escape of steam, lubricating arrangements, light and air in the engine-room, sufficient strength without excessive weight in any of the parts of the engine; also the pounds of coal, per horse-power, per hour needed to run it.

It is difficult to point out any position of the general duties of an engineer of more importance than the classification of the machinery requisite for the hull of a vessel. How few consider what is due to those who, with untiring energy, devote time, money, and even health, to attain a given effect of mechanism. The mass, gazing on the complicated disposition of the component parts of a pair of marine engines, are merely mechanical observers. The position of a given lever or shaft, right or wrong, is all the same to them; doubtless, the thought rarely, if ever, enters their mind that the skill and fertility of the brain must have been well tested to produce the subject before them.

Marine engines may be divided into two classes — beam and direct-acting. These may be either condensing or non-condensing; the former, however, are the most extensively used. With the exception of small screw-propellers and tug-boats, and steamers on the Western rivers, the condensing or low-pressure engines are almost exclusively used in this country. What, however, would be considered in this country as low-pressure steam, say 40 pounds to the square inch, would be considered in England as high-pressure.



VERTICAL COMPOUND ENGINE.

COMPOUND ENGINES.

The compound engine is a high- and low-pressure condensing engine, having two ordinary steam-cylinders, the smaller or high-pressure cylinder communicating direct with the boiler; the larger or low-pressure condensing cylinder direct with the condenser, and both with each other. The steam is admitted freely from the boiler into the high-pressure cylinder until the piston has been moved through a certain distance, when the valve closes, and the remainder of the space to be passed through by the piston is performed by the expansion of the steam, which, after doing its work in the high-pressure cylinder, passes into the condensing cylinder, where it does a proportionate amount of work, and then escapes into the condenser.

The cylinders of an ordinary condensing engine are both in connection with the condenser, that is to say, both cylinders receive steam direct from the the boiler, and both exhaust into the condenser. But with the compound engine, one cylinder only receives steam from the boiler; and, instead of exhausting its steam into the condenser, it exhausts into a jacket, or into the low-pressure cylinder casing. And as the communication to the condenser can only be obtained through the low-pressure cylinder, it follows that the quantity of steam admitted into the high-pressure cylinder does a certain amount of work in it and then exhausts, and before reaching the condenser does a certain amount more of work in the low-pressure cylinder; and as the high-pressure cylinder's exhaust steam enters the low-pressure cylinder at a much reduced pressure, the low-pressure cylinder is made larger in diameter than the high-pressure, for the purpose of equalizing the power, and also producing the desired ratio of expansion.

When the length of stroke of both cylinders is the same, it has been found, from modern practice, that the area of the condensing cylinder should be about three times that of the high-pressure one, and this proportion is best suited when the steam employed is from 45 to 50 pounds pressure above the atmosphere, and cutting off the steam after being admitted during $\frac{1}{3}$ of the stroke in the high-pressure cylinder. When the steam to be employed is of a less pressure, but the point of cut-off the same, then the relative proportions of the cylinders must be nearer to each other, and the reverse, when steam of a greater pressure is to be used.

The superiority of compound engines over single cylinder engines is due to the fact that the difference of extreme temperatures in each cylinder is less, and that the interior condensation is much diminished; and also to the partial removal of the water from the steam which held it in suspension at the time of its leaving the first cylinder, so that the water does not pass, in a liquid form, at least, to the second cylinder.

The amount of resisting pressure against the high-pressure piston can only be determined by the use of the indicator, as it is due in part to the atmosphere, and also to the amount of expansion that the steam undergoes on being exhausted into the jacket or receiver; therefore, the larger the area of the jacket, or receiver, the greater is the amount of expansion and consequent diminution of pressure, or back pressure, upon the high-pressure piston. But this back pressure is of importance in a compound engine, as it equalizes the power developed by both cylinders, inasmuch as the greater the back pressure on the high pressure piston, the less will be the power developed by the high-pressure engine, but the greater power will be developed by the low-pressure en-

gine; and should the high-pressure engine exhaust at or under the atmospheric line, the greater will be its power; consequently, the less will be the power of the low-pressure engine.

Therefore, in compound engines, the area of the jacket, or receiver, should be such as when the high-pressure steam is cut off at a certain portion of the stroke and exhausted into this jacket, or receiver. The pressure of steam that will enter the low-pressure cylinder shall be such as will develop in it equal, or nearly so, power to the high-pressure cylinder.

As an expansive engine, the compound engine has produced very economical results in many instances; yet it is almost too soon to speak with any degree of certainty as to its general introduction for marine and stationary purposes, as there are questions involved which will require several years to determine. Even if compound engines should be found in all cases capable of producing all the economical results claimed for them by their advocates, they would still labor under the great disadvantages of increased first cost, excessive weight, and complication of parts.

The compound engine, both for marine and stationary purposes, has had the position of its cylinders and the combinations of its parts arranged in many different ways, in some cases to suit the space available for its erection, and in others according to the different ideas of the different manufacturers; but the principle being the same in all cases, an equal economy should be obtained, if care is taken in so proportioning the passages for the steam that no undue obstruction is caused, and that proper and efficient means are employed to prevent any waste of heat.

The principle of the compound engine was known as early as 1781, when Hornblower obtained a patent for em-

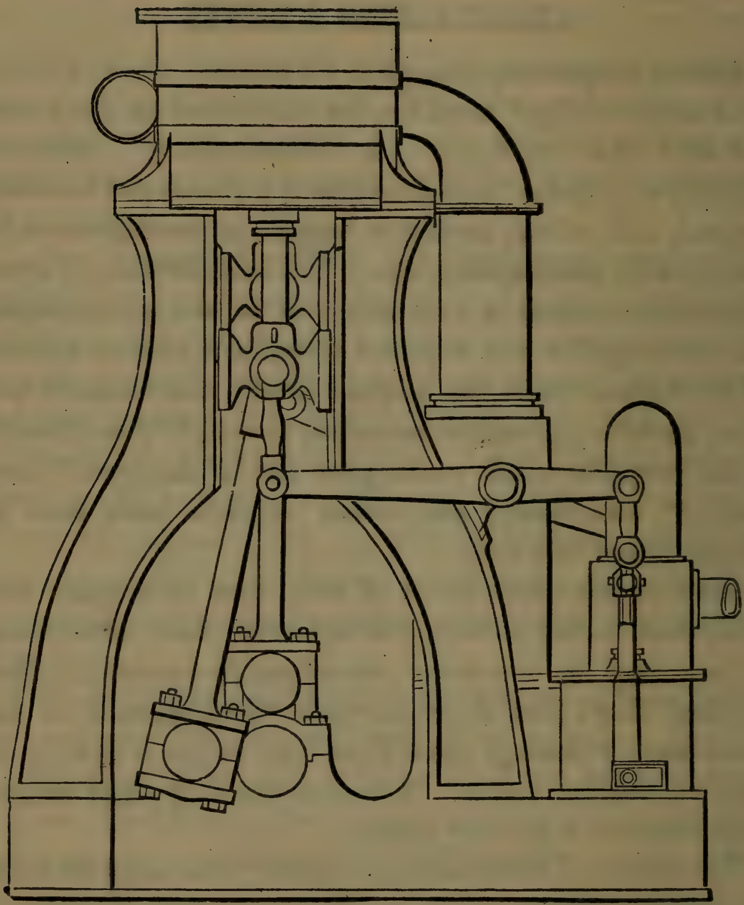
ploying steam, after it had acted on one piston, to operate on a second by allowing it to expand itself. But Hornblower was never able to practically apply the principle, which was successfully carried out by Wolf in 1804.

DIRECT-ACTING ENGINES.

Among engineers, generally, the phrase "direct acting" is an acknowledged term for the connection of the piston-rod with the crank-pin. A "direct-acting" engine is actually a cylinder having within it a piston-rod centrally secured, and of the requisite length, proportionate to the stroke, with stuffing-box, etc. The introduction of screw propulsion created a new exigency in steam mechanics. As the propeller was required to make a greater number of revolutions than the engines could conveniently perform, therefore it became necessary to couple them directly to the propeller shaft, which gave rise to the term "direct acting." Oscillating and trunk engines have been arranged under this title.

Most of the early forms of this class of engines were crude and unsatisfactory, consequently their introduction at first had to encounter considerable opposition from nautical men; but their excellent performances of late years have redeemed them from the disgrace that at one time seemed inevitable. They are less bulky and weighty than beam or side-lever engines.

The stroke of direct-acting engines, from want of room, is generally very short; but when it becomes necessary to give a direct-acting engine a long stroke, the object is effected by inclining the cylinder. Such an engine is then termed an inclined engine.



DIRECT-ACTING SCREW-ENGINE.

BALANCING THE MOMENTUM OF DIRECT-ACTING ENGINES.

The application of balance - weights to the cranks of direct-acting screw-engines is now a very general practice; and it is found to conduce to the easy and steady working of the engines in a very marked manner.

The principle on which the size of the counter-weights should be adjusted to the wants of the engine is of very easy apprehension. If the centre of gyration of the counter-weights describes a circle of the same radius as that described by the crank-pin, then the counter-weights must just be as heavy as the piston, and all the parts which move with it.

But if the centre of gyration of the counter-weights has a greater radius than the crank-pin, the counter-weights must weigh less than the piston and its connections, and if it has a less radius, they must weigh more. The only material condition being that the momentum, or amount of mechanical power resident in the counter-weight when moving in one direction, shall balance the momentum of the piston and its connections when moving in the opposite direction, and which weight may be supposed to be collected in the crank-pin.

OSCILLATING ENGINES.

Oscillating engines are a class of direct-acting engines which have no connecting-rod, but in which the piston-rod takes direct hold of the crank-pin; consequently, the cylinder oscillates or vibrates on trunnions, set near its centre, and through which the steam passes and the exhaust escapes. These engines have been successfully applied to paddle-wheel and even screw propulsion.

The objections formerly raised against their employment for marine purposes, namely, that the vibration of the cylinder would become a formidable evil in the case of a vessel rolling heavily at sea, and that the eduction passages being more tortuous than in common engines, the steam would escape less freely, and consequently induce back pressure, and that it would be difficult in large oscillating engines to obtain sufficient surface of trunnions to prevent them from heating, have proved entirely hypothetical, as they have all been successfully overcome by a judicious arrangement of the parts.

It is desirable to make the ends of the cylinders of ordinary engines as thin, relatively, as is consistent with the proper strength of the parts and the provision for a suitable stuffing-box. But with oscillating engines the case is different, as the strains developed by the oscillating motion of the heavy cylinder induce a great amount of side pressure in alternate directions on the piston-rod, at the point where it passes through the cylinder cover. Consequently, the covers of oscillating cylinders are necessarily deeper at their central portions than the like parts of other engines. The accurate fitting of this part of an oscillating engine is a matter of great importance.

The piston of an oscillating engine must play tightly and easily through the hole in the cylinder-head, while provision must also be made for receiving the lateral strain due to the oscillation of the cylinder. This is effected by lining the aperture for a considerable part of its length with brass, and making it in another portion an ordinary stuffing-box. It is also necessary that the piston-rod and crank-pin shaft should have larger proportions than those of other engines. The modes of attaching, or rather suspending, the cylinders of oscillating

engines differ somewhat. For although a vertical position is to be preferred on account of the weight of the cylinder appendages, etc.; yet an angular position is often adopted in consequence of being more available for long strokes and shallow vessels.

Oscillating engines have the advantage of simplicity of design and fewness of parts, and, in consequence of their diminished rubbing surfaces, they require less attention and less oil than any other class of engines, and all their principal parts are readily accessible should any defect be discovered.

TRUNK-ENGINES.

The trunk-engine has no piston-rod, but is a direct-action engine, in which the connecting-rod takes hold immediately on the piston itself, through a hollow, open cylinder within the steam-cylinder. This engine is favorable on the score of room, but increases the size and weight of the cylinder in order to obtain a given surface of piston on which the steam acts.

With the trunk-engine, the force exerted by the steam against the piston is by no means concentrated at once, but rather expended at right angles from the face of the piston to the crank-pin. This class of engine admits of a longer connecting-rod than some other forms of the direct-acting engine, and is composed of fewer parts. But it is very wasteful of steam, as the large mass of metal of the trunk, moving alternately into the atmosphere and cylinder, must condense a portion of the steam. The trunk also requires a large amount of packing, oil, and tallow.

GEARED ENGINES.

Geared engines are a class of engines in which, to obtain an increased number of revolutions of the pro-

peller without an increase in the speed of the piston, gear-wheels are introduced, by which the motion can be multiplied to any required number of revolutions of the screw. Formerly, a piston speed to produce 80 or 100 revolutions of the screw was considered impracticable, even in short-stroke engines, consequently, it was the universal custom to obtain it by the intervention of multiplying cog-wheels. But with the modern direct-acting engines any desired number of revolutions of the screw can be obtained without the employment of gearing.

The disadvantages of geared engines are that they are larger, heavier, and occupy more space than direct-acting engines; and, as a result, they fell into disfavor and had nearly gone out of use; but of late years their number appear to have increased, as many engineers prefer gearing to the high piston speed required in the direct-acting engine. The wear on the air-pump is also less with geared engines than with direct-acting engines, as its action is more uniform and moderate.

BACK-ACTION ENGINES.

The back-action engine is one in which, for economy of room and to lengthen the connecting-rod, the cross-head and the cylinder are on opposite sides of the shaft. This involves the necessity of a double piston-rod to a single cross-head, from which a single connecting-rod works back on the crank.

The steeple-engine may be said to be a form of back-action engine, with only one piston-rod, strapped to the middle of the shortest side of a triangular or harp-shaped iron frame, within which the crank revolves, and from the apex of which a pair of connecting-rods work back on the crank.

SIDE-LEVER ENGINES.

The side-lever engine is a modification of the beam-engine. In river and coast boats the working-beam, or lever, is above the engine, and single; but in the sea-going steamers, two of these beams are used instead of one; and instead of being above the engine, they are brought down to the bottom, one on each side, and being connected by a cross-tail, they act as a single beam or lever. Hence is derived the name from the disposition of the working-beam, the "side-lever engine."

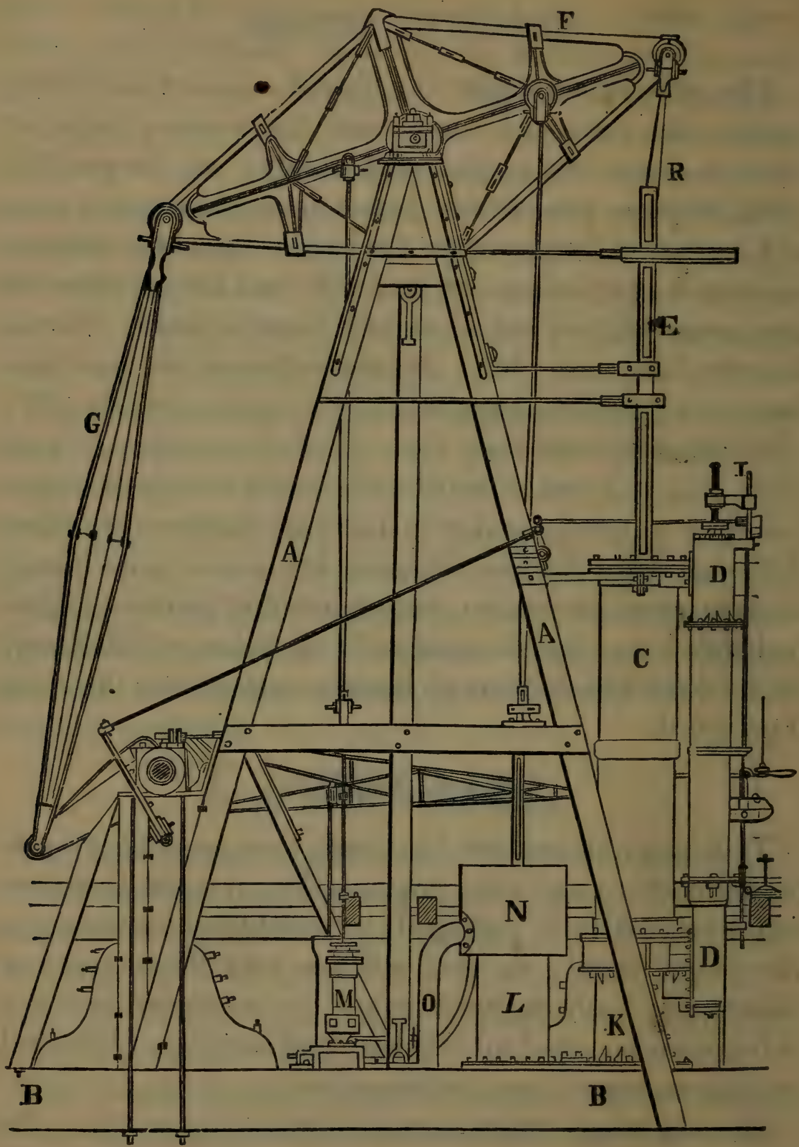
So universal has been the use of this engine in past years, that, to speak of marine engines, this form of engine would be ordinarily understood, unless otherwise specified. Although other designs of engines are now of more importance to steam navigation, certain it is that no other engine in its day was found equal to it in point of efficiency, its great drawback being its immense weight and the room it occupied.

BEAM-ENGINES.

This class of engines has been very successfully employed on coast and river boats, and even ocean steamers, and when skilfully managed is capable of attaining a very high speed; in fact, when a very high speed is desired, the beam-engine is generally employed, as it is a well-known fact that all other engines have been distanced by their extraordinary performances.

The objections urged against the employment of the beam-engine for marine purposes, namely, its great oscillation, the resistance due to the wind, exposure in time of war, etc., have not been sufficient to counterbalance, or even equal, its other good qualities.

Beam-engines also present great facilities for procuring



MARINE BEAM-ENGINE.

A, A. Main frame; B, B. Keelsons; C. Cylinder; D, D. Valve-chest; E. Guides; F. Beam; G. Connecting-rod; I. Valve-stem; K. Condenser; L. Air-pump; M. Boiler feed-pump; N. Hot-well; O. Delivery-pipe; R. Link-rod, or main link.

every possible variety of reciprocal motion for pumps of less stroke than the piston, as the various feed-pumps, bilge-pumps, and air-pumps are very conveniently attached to the beam at any point from the centre, outward, and each, consequently, is moved with a length of stroke proportional to such position.

Beam- and side-lever engines differ from direct-acting engines simply in the method of taking the power from the piston-rod. In the one, the piston-rod is connected directly with the crank; while in the other, the working-beam, vibrating on its centre, receives at one end the power from the piston-rod.

MARINE BEAM-ENGINE.

The cut on the opposite page represents a marine beam-engine. The framework, it will be observed, is composed of four pieces of wood, which are formed into two triangles inclined laterally to each other. They are fastened together, and to the boat, by numerous horizontal and diagonal timbers, which are secured by wooden knees and keys. The two front legs of the framing are bolted and keyed to diagonal flanges cast on the sides of the condenser. At the other end, the framing is attached to the timbers which support the shaft pillow-blocks. The framing is further steadied by two additional timbers running from the beam pillow-blocks outside the shaft to the keelsons. The entire fastening of the engine and framing is so arranged as to reduce all the strains to direct ones of extension or compression of the fibres of the iron and wood employed in the construction.

The working-beam is composed of a skeleton frame of cast-iron, round which a wrought-iron strap of great strength is fixed. This strap is forged in one piece, and

its extreme ends are formed into large eyes, which are bored out to receive the end-journals. The skeleton frame is a single casting in the form of a cross, and contains the eyes for the main centre and air-pump journal.

The bed-plate is a single casting, and forms the foundation of the heavier portions of the engine. It is carefully fitted upon the keelsons of the boat, and is firmly secured by numerous holding-down bolts. That part of the plate which lies between the keelsons forms the channel-way or passage from the condenser to the air-pump. In the centre of this passage are foot-valves.

The condenser is of a cylindrical form, flanged at both ends, and of the same diameter as the steam-cylinder; its contents are about $\frac{13}{30}$ths of that of the space through which the piston passes during one stroke. The upper extremity is cast close; the lower end is open, and is fitted down to the chipping fillets on the bed-plate, to which it is firmly bolted and secured by a rust-joint. On the sides of the condenser, and running in an inclined direction, strongly bracketed webs or flanges are cast, to which the wooden framing that supports the main beam is fastened by bolts and keys.

The cylinder bottom is a circular-flanged casting containing the lower steam-port. It forms the connection between the cylinder and the condenser, to both of which it is fitted and bolted. The steam-cylinder is secured to its bottom by a rust-joint. It stands vertically over the condenser, and has its upper end steadied by horizontal stays to the framing.

The piston is of the ordinary form of spring packing, except that, in consequence of its great area, it has to be strengthened by radiating arms cast on the top and bottom flanges. The cylinder cover is ribbed on the inner side similar to the piston-head, its upper or outer surface being turned and polished.

The steam-chests contain the valves and seats and the inlet and outlet steam passages. On the upper chest is cast the throttle-valve pipe, to which is attached the supply-pipe leading from the boilers. On the bottom chest the exhaust branch is cast, through which the waste steam passes to the condenser. The valve-bonnets and glands are turned and polished. The chests are ground-jointed to the upper and lower steam-ports of the cylinder.

The side-pipes, which connect the steam-chests, are of cast-iron, ornamented with bands and mouldings, and turned and polished throughout their entire length. At the upper end of each pipe is an expansion ring of thin copper, which, by its yielding, compensates for any slight elongation or contraction of the side-pipes occasioned by heating and cooking.

The valves which govern the entrance and exit of the steam are connected together in pairs, and of the kind called double balance-valves, from the fact that the downward pressure on one valve is balanced, or nearly so, by the upward pressure on the other. The upper valve of each pair on the steam-side, and the lower one of each pair of the exhaust, are a little larger than the others; and, consequently, there exists a small amount of unbalanced pressure, which effectually retains the valves in their seats.

The valve-gear consists of the lifter-rods with their lifters, and the rock-shafts with their levers. There are four lifter-rods, which are polished bars of wrought-iron, placed in front of the steam-chests. They are made to move vertically up and down through guides which are cast or bolted to the chests and side-pipes. On the lifter-rods are keyed eight projecting arms, called lifters. Four of these embrace the extremities of the valve-spindles, which are screwed, and provided with double jam-nuts.

The remaining four lifters are likewise keyed upon the

rods, and are placed directly over the levers on the rock-shafts, from which they receive their motion. There are two rock-shafts—one for the steam, and one for the exhaust-valves, which are worked by separate eccentrics. On the shafts there are four levers, by which the lifters and rods are raised, and they are curved on their working faces, in order to render their action smooth and noiseless.

By the reciprocating or rocking motion of the shafts, the lifter-rods, and with them the valves, are alternately raised and lowered. The exhaust-valve levers are of a length just sufficient to give the requisite amount of lift and lead, and they are so adjusted on their rock-shaft that the moment one rod is fairly down, the raising of the other commences.

The steam-levers are considerably longer, and are placed upon their rock-shaft in a position inclined to one another, so that an interval, longer or shorter, occurs between the falling of one rod and the raising of the other. During this interval both valves are down, and the steam is, of course, shut off from the piston. This arrangement constitutes the expansive cut-off gear, and it may be varied by altering the position of the eccentrics on the shaft, the levers on the rock-shaft, and the pin in the eccentric lever. The amount of lift of the valves may be regulated by moving the eccentric-pin.

The hand rock-shaft, or trip-shaft, is a small shaft of wrought-iron working in bearings cast on the lower steam-chest. It has solid projections upon it corresponding to similar ones on the lifter-rods, and its reciprocating motion raises and lowers the valves in precisely the same manner as the large rock-shafts. Sockets are formed in the trip-shaft, into which the starting-bar is inserted. The leverage of this is considerable, whilst the resistance amounts to but little more than the weight of the lifter-

rods, valves, and their appendages; consequently, the handling of the engine is performed with great facility.

STARTING-GEAR FOR MARINE ENGINES.

The form of starting-gear for marine engines is usually a wheel with handles at certain distances on the periphery of its rim; this wheel, being keyed on the end of a shaft, having a worm at its opposite extremity, which worm gives motion to a toothed segment keyed on a weigh-shaft centrally, at each end of which are levers, connected by a rod to the slide-valve link.

Another arrangement of starting-gear is a wheel and shaft as above, having keyed on the weigh-shaft a spur-pinion, which imparts motion to a spur segment; this motion being conveyed to the link as before stated. This last arrangement is of a more simple character than the former; but the spur-gear necessitates a friction stop on the hand-wheel shaft, to prevent the latter from turning during the motion of the engines.

Perhaps the most modern arrangement is a mitre-wheel keyed on the starting-wheel shaft; the former gears with another on the end of a rod, having a coarsely pitched screw chased on it; the screw works in the boss of the last mitre-wheel, which revolves on the hand wheel giving motion to it, consequently causing the ascent or descent of the screw-rod, which is connected to the valve-link in the usual manner. In some cases the link-rod is connected to a sliding-block, which receives motion from the screw; the rod revolving thus gives motion to the link. This latter arrangement is mostly used for small engines.

In the larger class of marine engines, the links are now very generally moved by means of a separate engine, to

the cross-head of which they are attached by means of rods.

CONDENSERS.

In the condensing engine, when the steam is exhausted from the cylinder, it escapes to the condenser, where it is, as the name implies, condensed into water by being brought in contact with a jet of cold water, or by passing through a series of tubes, over and around which a stream of cold water is continually passing. The former method is what is known as "jet condensing," while the latter is what is termed "surface condensing."

When the jet condenser is used, salt water must be pumped into the boilers, as the water of condensation and that of the jet mingle; but the surface condenser, if perfectly tight, saves the water of condensation, and it can be returned to the boilers again and again; by this means nearly all the feed-water is fresh, and but little blowing off is required to keep the water at a low point of saturation.

The advantages claimed for the surface condenser over the jet are, that it furnishes marine boilers with distilled instead of sea-water, which must make a great saving in fuel, as it obviates the necessity of continually blowing off a portion of the water to keep the saturation at a desired point, and that it prevents, to a certain extent, the loss and danger incurred by the accumulation of salt and scale on the heating surfaces of the boilers, and also lessens the expense of cleaning and repairing.

But in practice it has been found that the gain by the use of the surface condenser is not nearly so large as theory would indicate. Nearly all of them leak to some extent, so that salt water mingles with the fresh water of condensation. Moreover, all the water that is evaporated by the boilers is not preserved in the condenser, so that salt feed has to be used sometimes.

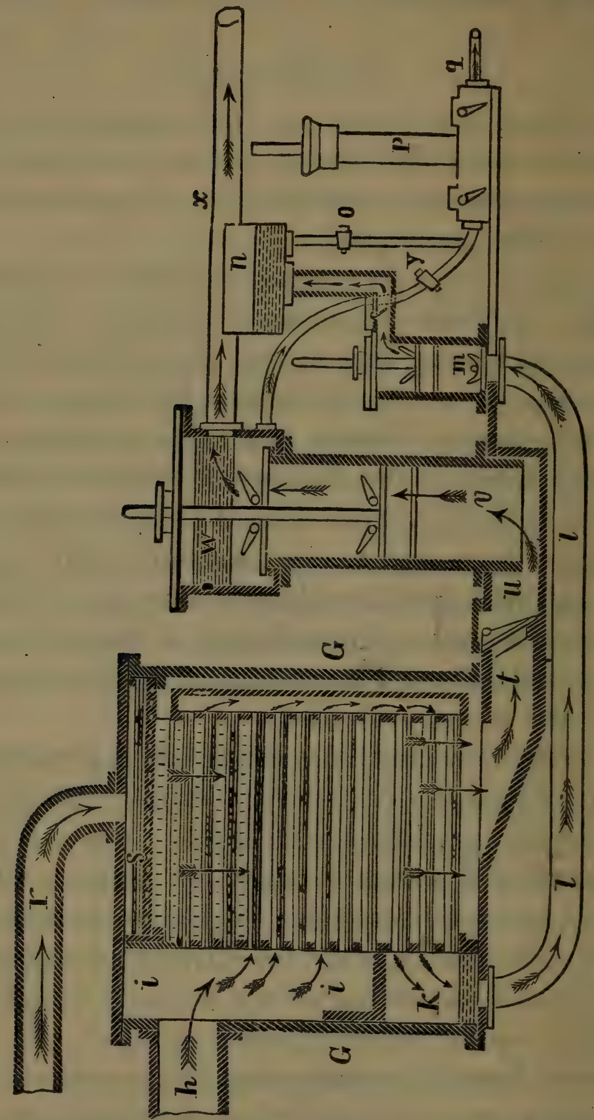
Surface condensers are much heavier, more expensive, and occupy more space than the jet condenser. The vacuum also is not so good as that produced by the latter. It has also been found that surface condensers induce corrosion in the boilers; but even with all these disadvantages, it must be admitted that it is a most important adjunct of the marine engine.

The cut on page 218 represents the construction and operation of Pirsson's surface condenser. The engine being put in motion, the exhaust steam flows through the exhaust-pipe, *h*, into the chamber, *i, i;* thence in the direction of the arrows through the tubes, returning through the lower tubes to the chamber, *k;* injection water being admitted at the same time from the sea through the injection-pipe, *r*, is showered by the scattering-plate, *s*, over the tubes, and by its gravity takes the direction of the arrows to the channel-way, *t*, from which it is removed by the air-pump, *v*, and delivered into the hot-well, *w*, to the delivery-pipe, *x*, and thence overboard.

The water resulting from condensation is drawn by the fresh-water pump, *m*, from the chamber, *k*, through the pipe, *l, l*, and delivered into the fresh-water reservoir, *n*; from this reservoir it passes through the pipe, *o*, to the feed-pump, P, and is delivered into the boilers through the feed-pipe, *q*. The pipe, *y*, is for the purpose of supplying salt water to the feed-pump, P, when there is a deficiency of fresh water in the reservoir, *n*.

In some forms of condensers, the operation is the reverse of the one just described, as the exhaust steam is received on the outside of the tubes, and is condensed by water circulating through them.

The quantity of injection water and water of condensation is about the same with both surface and jet condensers. Therefore, the work performed by the single air-



PIRSSON'S SURFACE CONDENSER.

Explanation. — G, G, condenser; *h*, exhaust-pipe; *i*, *i*, exhaust-chamber; *k*, exhaust-passage; *l*, *l*, fresh-water pipe; *r*, injection-pipe; *s*, scattering-plate; *t*, injection channel; *v*, air-pump; *w*, hot-well; *x*, delivery-pipe; *m*, fresh-water pump; *n*, fresh-water reservoir; *o*, fresh-water supply-pipe; P, feed-pump; *q*, feed-pipe; *y*, salt-water supply-pipe.

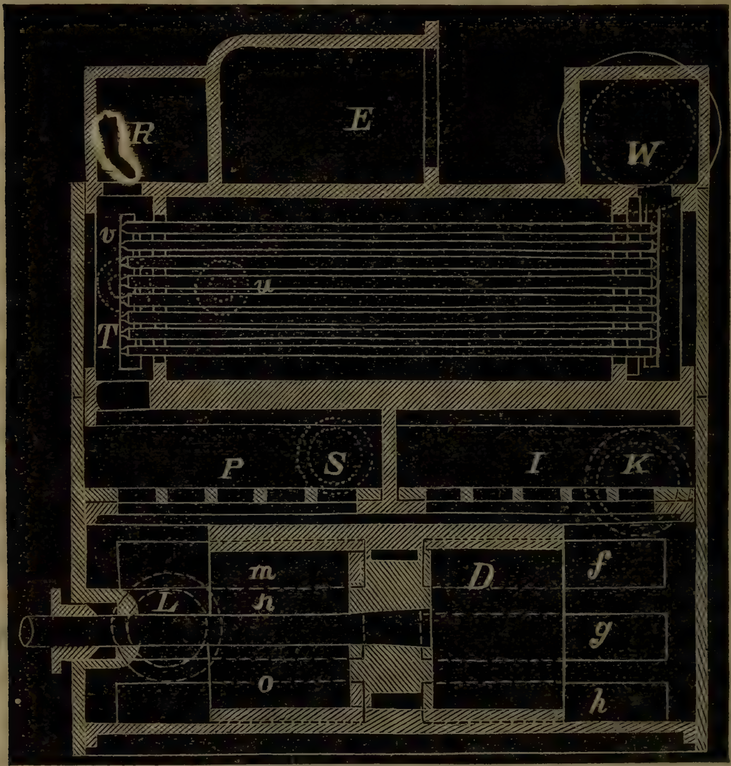
pump of the latter is the same as that performed by the two air-pumps of the former. The temperature of the feed-water, in both cases, ranges from 100° to 110°; so that it is obvious that the gain with the surface condenser is reduced to that due to not blowing out, and to the cleaner condition of the boilers, as stated in a former paragraph.

When the engine is standing still, it frequently happens that, in consequence of leaky steam- and exhaust-valves, the condenser becomes too hot; consequently, when it is necessary to start, the pressure is so great in the condenser that the injection water will not enter. Overheating of the condenser is always indicated by a cracking noise. In such cases, it is always best to pump some cold water into the condenser, if there be any independent arrangement for that purpose; if not, the temperature can be lowered by pouring cold water over the outside of it, or the pressure can be lessened by moving the engine back and forth two or three revolutions.

Relative Quantities of Injection and Condensed Water.—The injection water enters the condenser at a certain temperature, and, coming in contact with the steam, its temperature is increased and that of the steam diminished, all the latent heat of the steam being made sensible. It only becomes necessary to ascertain the quantity of water that will absorb the heat contained in the steam.

Suppose the temperature of the steam entering the condenser to be 1188°, and that of the injection water 60°, while the discharge water is 110°. Now, as the water of condensation has a temperature of 110°, there are $1188° - 110° = 1078°$ imparted to the injection water. But a quantity of injection water equal to the water of condensation receives $110° - 60° = 50°$. Hence the quantity that receives 1078° of heat will be $1078° \div 50° = 21{\cdot}56$ times

the water of condensation; or, in other words, the injection water is necessarily from twenty-one to twenty-two times greater than the water of condensation.



Sewell's Surface Condenser.

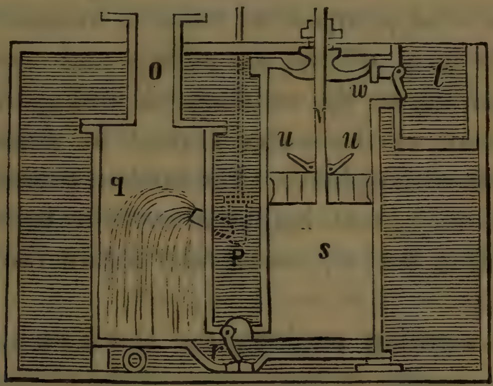
The above cut represents Sewell's surface condenser. The air-pump, D, is double-acting, and has two sets of foot- and delivery-valves — one set being for the injection, and the other for the water of condensation. The injection water enters at the opening, L; is drawn through the foot-valves, M, N, O, and forced through the delivery-valves, P, through the tubes, T, and overboard through the overboard delivery, W. The exhaust steam enters at E, and is condensed by contact with the outer surfaces of the tubes.

The water of condensation is drawn through the foot-valves, F G H, and is forced through the delivery-valves, I, and the outboard, K, into a reservoir, from which the feed-pumps draw their water.

There is a loaded valve in this reservoir, communicating with the outboard delivery, so that, when the reservoir becomes full, the water may escape. The openings, V U, are the ends of a pipe connecting the fresh- and salt-water reservoirs, so that any deficiency in the feed-water may be supplied from the latter reservoir. R is an air-chamber for the salt-water reservoir, and S is the end of a pipe through which the auxiliary pump draws water.

The tubes in this condenser are not secured to the tube-heads, but pass through them, and are made tight by rubber grummets. By this means each tube is allowed to expand and contract independently of all the others.

The following cut represents a jet condenser.



Jet Condenser.

O, Exhaust-pipe; P, injection-pipe; Q, jet; R, foot-valve; S, air-pump cylinder; U U, air-pump valves; V, air-pump rod; W, delivery-valve; T, hot-well.

AIR-PUMPS.

All condensing engines have, of necessity, to be supplied with an air-pump, the chief function of which is to remove the injection water, or water of condensation, and extract the air that has been liberated from the water in boiling, from the condenser.

The proper proportions of the air-pump were determined by Watt to be about one-eighth of the capacity of the cylinder. In more modern engines, especially where there are irregularities of motion, the air-pump is generally made a little larger than this proportion.

The capacity of the air-pump depends on the quantity of water injected, and this again depends on the quantity of steam to be condensed. If the steam is worked expansively, the pump may be smaller than if it was worked whole-stroke. And if it be double-acting, it need only be half as large as if single-acting.

If the condenser be considered as a well from which the water is to be pumped, we notice a manifest difference between the duty of the air-pumps and that of other pumps, inasmuch as the condenser is not open to the air, and the ascent of the water cannot be assisted by the pressure of the atmosphere. The bucket, therefore, cannot be raised to any great height above the condenser with the expectation that the water will follow the bucket on its up-stroke.

In quick-working engines, the shock on the delivery-valve, on the ascent of the bucket of the air-pump, is considerable, and produces a very disagreeable noise. To prevent this, a small valve is sometimes fitted, to admit air above the bucket as it descends.

The air-pump is generally attached to the sole-plate by a faucet-joint, which is preferable to a rust flange-joint, as the salt water eats away the heads of the bolts, unless

they are copper; and if they are copper, they waste the iron. The oil and grease which fall from the machinery upon the sole-plate deoxidize the rust of a flange-joint; whereas with a faucet-joint, suitably made, they cannot remain in the same intimate contact. Air-pump barrels, buckets, and valves are almost entirely made of brass or Muntz's metal; the rods are generally made of iron and covered with a skin of brass; the valves are most commonly of the spindle or pot-lead kind.

The most common methods for working air-pumps are either by a rod from the main beam, or by the oscillation of the cylinder or a crank or large eccentric on the main shaft; although air-pumps are in some cases worked independently by means of a separate engine.

THE HYDROMETER, SALINOMETER, OR SALT-GAUGE.

Ocean steamers, using sea-water in the boilers, require frequent change of the water to prevent incrustation, or deposit of salt and earthy matter upon the flues and within the legs of the boilers. This exchange should be made with regularity and care, lest, on one hand, the object sought should not be attained, or, on the other, a waste of heat should be occasioned in discharging hot water too freely from the boiler, the place of which is to be supplied with cold.

Salt-gauge.

It is, however, better, as a general thing, to err on the side of a too liberal use of the blow-off cock; for the loss of heat would probably be less from this cause than it would be if the boiler were allowed to become incrusted with a non-conducting substance. At what exact degree of saturation saline incrustation begins has not as yet been

determined, but it appears to vary considerably under different circumstances; consequently, the wisest course to pursue would be to be governed by practice rather than by theory.

It is necessary, therefore, to have some test by which the saltness of the water may be known, and, having this, to adopt such a system of blowing-off as will keep the water uniformly at the degree fixed upon. The degree of saltness is ascertained by the hydrometer. It is graduated to show the number of ounces of salt contained in 1 U. S. gallon (8·338 avoirdupois pounds), and can be made of either glass or metal.

Sea-water is generally understood to contain $\frac{1}{33}$, by weight, of salt; but this proportion varies in the water of different seas, as will be seen in the following table:

	Parts in 1000.			Parts in 1000.	
Baltic Sea,	6·60	$= \frac{1}{152}$	Mediterranean Sea,	39·40	$= \frac{1}{25}$
Black Sea,	21·60	$= \frac{1}{46}$	Atlantic at the Equator,	39·42	$= \frac{1}{25}$
Arctic Sea,	28·30	$= \frac{1}{35}$	South Atlantic,	41·20	$= \frac{1}{24}$
Irish Sea,	33·76	$= \frac{1}{30}$	North Atlantic,	42·60	$= \frac{1}{23}$
British Channel,	35·50	$= \frac{1}{28}$	Dead Sea,	385·00	$= \frac{11}{28}$

The boilers of ocean steamers are usually filled with fresh water before starting on their voyages, and, as they are generally fitted with "surface condensers," they use no salt water at all, the loss of fresh water induced by leakage, etc., being replaced by water distilled by an apparatus with which nearly every ocean steamer is supplied.

THE MANOMETER.

The Manometer, according to the derivation of the word (from *manos*, rare, and *metron*, measure), is an instrument for measuring the degree of rarefaction of aeriform fluids subjected to less than atmospheric pressure.

The term is also, and more generally, applied to the instrument when used to indicate the density of aëriform fluids subjected to more than atmospheric pressure. The manometer may therefore be defined to be an instrument for measuring the density of aëriform fluids by means of a glass tube inserted in a reservoir of mercury. At the ordinary pressure of the atmosphere, the mercury will stand about 29° or 30° above the level of the mercury in the reservoir; but as soon as the pressure above the surface of the mercury in the reservoir becomes less than that of the atmosphere, the column of mercury will, of course, fall in the tube just in proportion to the diminution of the pressure.

THE BAROMETER.

The Barometer is an instrument used for observing the pressure and elasticity, or variations in density, of the atmosphere. It is commonly employed for the purpose of determining approaching variations in the weather, and, more scientifically, for measuring altitudes. There are various modifications of the barometer, such as the diagonal, horizontal, marine, pendant reduced, and wheel barometer; in all of which the principle is the same, the only difference being in its application. The essential part of a barometer is a well-formed glass tube, closed at one end, perfectly clear and free from flaws, 33 or 34 inches long, of equal bore, filled with pure mercury, and inverted; the open end being inserted in a cup partly filled with the same metal, so that the mercury in the tube may be supported by atmospheric pressure.

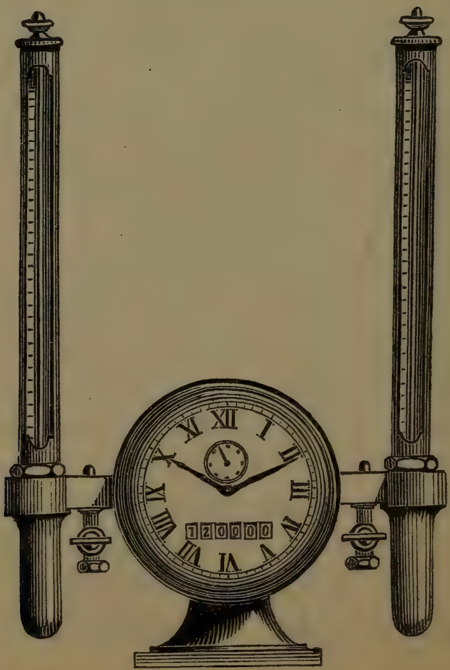
The excellence of the Barometer depends chiefly on the absence of all matter except mercury from the tube, and its value may be tested by three indications: — *First*, by the brightness of the mercurial column, and the ab-

sence of any flaw, speck, or dulness of surface; *secondly*, by the "barometric light," as it is called, or flashes of electric light in the Torricellian vacuum, produced by the friction of the mercury against the glass when the column is made to oscillate through an inch or two in the dark; *thirdly*, by a peculiar clicking sound produced when the mercury is made to strike the top of the tube. If air be present in the tube, it will form a cushion at the top, and prevent or greatly modify this click.

The vacant space between the top of the mercury and the top of the tube is called the Torricellian vacuum, in honor of the inventor of the instrument.

MARINE ENGINE REGISTER, CLOCK, AND VACUUM GAUGES.

The annexed cut represents a marine engine register or counter, clock, and steam- and vacuum-gauges. It con-



Marine Engine Register, with Clock, Steam-gauge and Vacuum-gauge.

sists of a circular cast-iron box, in which are cut, side by side, six (or more as may be required) slots, through which may be seen the numbers representing the revolutions of the engine; this is denominated the "counter" or register. By an attachment to any suitable part of the engine, a vibratory motion is communicated to an arm attached to a central horizontal shaft, placed parallel to the dial, and within the cast-iron box; to the ends of which is also fixed a frame carrying a small shaft parallel to the former, on which six palls, or arms, are attached, side by side, and at a certain distance apart, in such a way that the right-hand pall may fall without the others, but cannot rise without carrying all the rest.

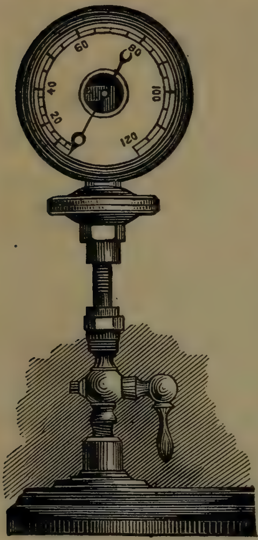
This framework, with the pall-shaft, etc., is made, by the motion of the arm attached to the engine, to describe an arc of 36°, or to move through $\frac{1}{10}$ of a circle. The ends of the palls respectively rest on and slide over six cylinders, placed side by side on the central shaft, all of which are free to move in the same direction and independently of each other, and are arranged as 1, 2, 3, 4, etc., beginning with the right-hand one.

On the right-hand edge of each cylinder are cut ten slots, and on the left-hand, which overlaps the edge of the next, only one slot — these slots being of such a size as will admit of one of the palls; then, on the back motion of the framework, etc., the pall is carried back till it drops in, when the forward motion carries with it the cylinder so locked.

STEAM-GAUGES.

The use of the steam-gauge is to indicate the steam pressure in the boiler, in order that it may not be increased far above that at which the boiler was originally consid-

ered safe; and it is as a provision against this contingency that a really good gauge is a necessity where steam is employed, for no guide at all is vastly better than a false one. The most essential requisites of a good steam-gauge are, that it be accurately graduated, and that the material and workmanship be such that no sensible deterioration shall take place in the course of its ordinary use.



Steam-gauge.

The pecuniary loss arising from any considerable fluctuation of the pressure of steam has never been properly considered by the proprietors of engines. If steam be carried too high, the surplus will escape through the safety-valve, and all the fuel consumed to produce such excess is so much dead loss. On the other hand, if there be at any time too little steam, the engine will run too slow, and every lathe, loom, or other machine driven by it, will lose its speed just exactly in the same proportion, and of course its effective power.

A loss of one revolution in ten at once reduces the productive power of every machine driven by the engine ten per cent., and loses to the proprietor ten per cent. of the time of every workman employed to manage such machine. In short, the loss of one revolution in ten diminishes the productive capacity of the whole concern ten per cent. so long as such reduced rate continues; while the expenses of conducting the shop (rent, wages, insurance, etc.) all run on the same, as if everything was in full motion. A variation to this amount is a matter of frequent occurrence, and is, indeed, unavoidable, without proper instruments to enable the engineer to avoid it.

A very little reflection will satisfy any one that it must be a very small concern, indeed, in which a half-hour's continuance of it would not produce a result more than enough to defray the cost of a very expensive instrument to prevent it. If the engineer, to avoid this loss, keeps a surplus of steam constantly on hand, he is then constantly wasting the steam, and consequently fuel, and thus incurs another loss, which, though less alarming than the first, will yet be a serious one, and render any instrument most desirable which can prevent it.

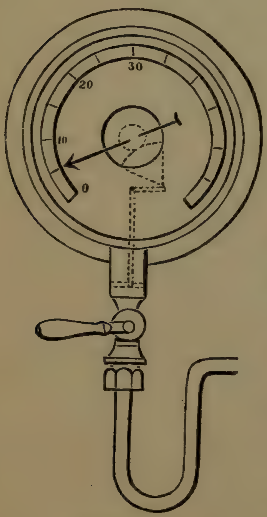
It is evidently, therefore, of great importance to the proprietors of engines to have an instrument which should constantly indicate the pressure in the steam-boilers. This would enable the engineer to keep his steam at a constant pressure, and thus avoid waste of fuel, on the one hand, and the still more serious loss of the productive power of the shop, on the other. An instrument, therefore, constantly indicating the pressure of steam, reliable in its character, and, with ordinary care, not subject to derangement, is evidently a desideratum both to the engineer and proprietor. The importance of such an instrument, as a preventive of explosion, and of the frightful consequences to life and limb, and ruinous pecuniary results of such disaster, is obvious on the slightest consideration; but the value of the instrument, in the economical results of its daily use, is by no means properly appreciated.

Steam-gauges are made in various forms, but they may be divided into two general classes—spring and mercury. The spring-gauge, in consequence of being cheaper, more compact and durable, and frequently more ornamental than the mercury-gauge, is more generally used on stationary and locomotive boilers; but it is less reliable than the latter—reliability being the great desideratum in a steam-gauge.

The Spring-gauge.—The principle of its construction is as follows: When a thin metallic tube is nearly flattened and afterwards coiled, the effect of any inward pressure is to force it towards its original shape,—the first effect produced being that of tension towards elongation, whether the flattened tube be coiled or twisted; and a contrary effect is produced by unresisted exterior pressure.

In shaping it, a certain degree of elasticity having been given to the metal, as long as it is not absolutely forced beyond a given point of its acquired shape, it will act as a spring to the greatest perfection, and work from or back to its newly acquired shape, as the pressure upon it may be applied.

Thus, a simple piece of well-made metal tube is first partially flattened in all its length, and coiled nearly to a circle. One end of it is stopped up, while the other is left open, to receive the pressure of steam or water. To the end that is stopped up a hand is fixed, which is so placed as to show the variations in the position of the tube upon a dial marking the degrees of pressure.



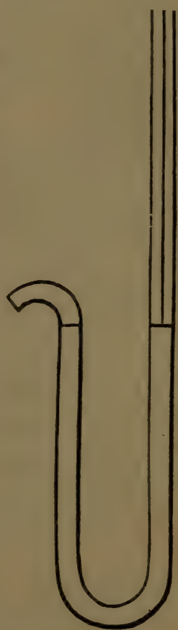
Vacuum-gauge.

The spring-gauge can also be used as a vacuum-gauge, by reversing the application of the pressure, which has a contrary effect on the tube. For instance, as exhaustion takes place in the tube, so does its power of resisting the pressure of the surrounding atmosphere which acts upon it vary, and it consequently again coils under that pressure in regular ratio with the variation of it, and is made to indicate the degree of vacuum in the condenser of an engine.

The Mercury-gauge.—The mercury-gauge is the surest and simplest of all gauges, but is often inconvenient on account of the space it requires. When used for stationary boilers, it consists of a vertical glass tube communicating with a cistern of mercury, which rests on a steel or gutta-percha disk; its chief drawback for stationary purposes is its first cost, which is about twice that of the spring-gauge. It is not adapted to locomotive boilers.

When used for marine purposes, it consists of a siphon tube partially filled with mercury, one end of which is subjected to the pressure of the steam, whilst the other is open to the air. The pressure tends to displace the column in one leg and raise it in the other; the difference between the two shows the amount of pressure.

An inverted siphon filled with mercury, with a rod floating on its surface, was among the earliest inventions of the fathers of the steam-engine; and for steam of one or two atmospheres, this is still the most reliable of the appliances in use at the present day; but for steam of higher pressure, they become less reliable and convenient, in consequence of their great height, the friction of the float, and their liability to lose the mercury by its oscillation or by excess of pressure.



Siphon-gauge.

The mercury, when not pressed upon by steam, will stand at a level in both legs of the siphon; and even when an atmosphere of steam has taken the place of the atmospheric air in the boiler, the mercury will still stand at the same level. But when the pressure increases, it will press with greater force on the mercury in the leg exposed to the pressure of the steam

than the atmosphere presses on the mercury in the leg open to the air; consequently, the mercury will rise in the open leg, and indicate the excess of pressure of the steam beyond that of the external air.

The graduations on the mercurial gauge are an inch in length. Every inch that the mercury rises in the gauge indicates an alteration of two inches in the level of the column supported in the bent tube; consequently, one inch of mercury on the scale indicates a pound pressure of steam.

A column of atmosphere 45 miles high, and one square inch in area, just balances, and consequently weighs the same as a column of mercury of like area and 30 inches high. This column of air also balances $33\frac{7}{8}$ feet of water. Consequently, a column of air 45 miles high, 30 inches of mercury, and $33\frac{7}{8}$ feet of water, weigh the same.

1 atmosphere, or 15 pounds per square inch, } = 30 inches of mercury.

Each pound pressure per square inch } = 2 inches of mercury.

Each pound pressure per square inch } = 1 inch rise on siphon-gauge.

1 atmosphere, 15 pounds per square inch, } = $33\frac{7}{8}$ feet of water.

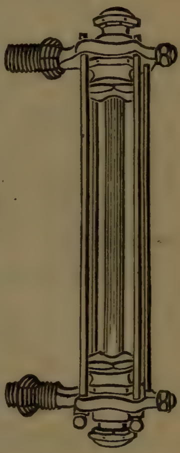
Each pound pressure per square inch } = $27\frac{1}{10}$ inches of water nearly.

Suppose it were possible to erect a tube having a sectional area of one square inch upon any part of the earth, and that this tube be long enough to reach up to the height of the atmosphere (which is supposed to be about 45 miles), the air contained in such a tube would weigh about $14\frac{3}{4}$ pounds. Now, if we take another tube, having

the same sectional area, and place 14¾ pounds of water in it, the level of the water will be found to be 33⅞ feet above the bottom of the tube. If we take still another tube of the same area, and place 14¾ pounds of mercury in it, the level of the mercury will stand 30 inches above its base.

GLASS WATER-GAUGES.

The glass water-gauge may be said to be one of the simplest, as well as one of the most useful, attachments of the steam-boiler, as by it the engineer can see at a glance the level of the water. No other method of determining the height of water in steam-boilers can be so reliable as a well-made glass water-gauge. Consequently, they have been almost universally used since their introduction to the present time.



It consists of a thick, well-annealed glass tube connected with two valves, the lower one of which enters the water, and the upper the steam space of the boiler, by means of which the level of the water is indicated directly in the tube. These valves can be opened or shut so that the tube may be filled with either water or steam, or both, or be blown out. The tube is packed on each end with thin rubber collars, which are made steam- and water-tight by means of stuffing-boxes.

A very important improvement has been recently made in glass water-gauges by M. Pennypacker, Supt. of the Baldwin Locomotive Works, Phila., by which the dangers heretofore incurred by the breaking of the glass tubes, in the absence of the engineer, is entirely obviated, as, in case the glass should become broken, the steam and

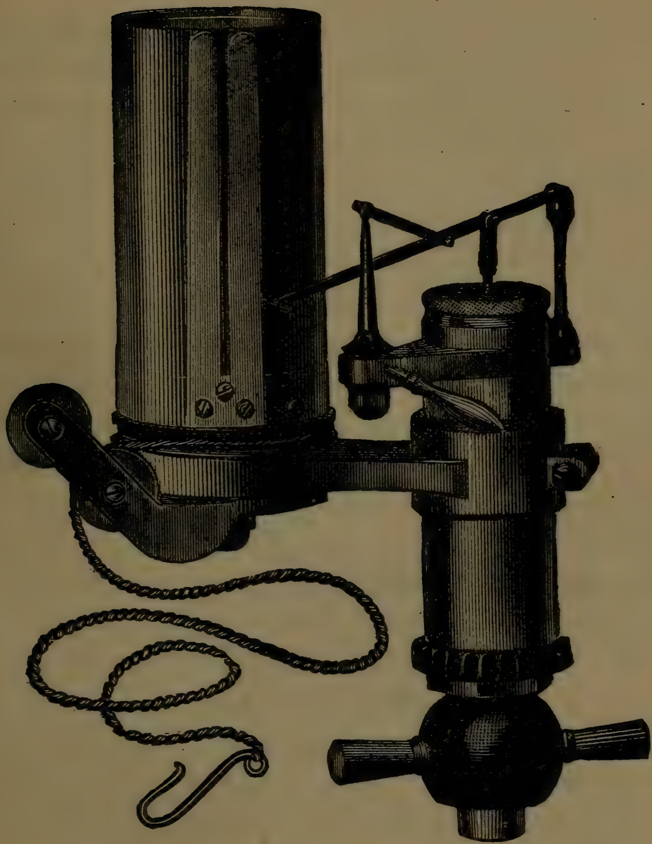
water communications with the boiler are instantly automatically closed by means of balanced valves.

As long as the glass tube is unbroken, the valves are nearly balanced, and will remain open, owing to their own weight; but should the glass tube break, the pressure on the top of the valves being removed, the pressure under them will instantly force them to their seats, thereby effectually shutting off the water and steam from the broken tube. This gauge is also so arranged that the steam and water may be shut off from the boiler in the usual manner, should the self-acting valves fail to close, as the position of the positive stop-valves, is placed between the self-acting valves and the boiler, which also allows the latter to be taken out and cleaned with the full pressure of steam on the boiler. These gauges will prove invaluable to steam-users, who are of necessity compelled to locate their boilers in the midst of valuable property.

Glass gauges require frequent blowing out, as the tube soon becomes discolored, and the lower or water connection is liable to become filled with mud. This can be done by opening the drip at the bottom of the gauge and closing the water-valve, when the steam will rush down through the tube and remove any deposit that may be on the inside of the glass.

Should it become necessary to use a swab, it must, in all cases, be of wood and covered with cloth, as the touch of any hard substance on the inside of the glass produces an immediate abrasion. If the end of the swab be dipped in acetic acid, it has the effect of removing all discoloration from the inside of the tube.

Magnetic water-gauges are sometimes used on stationary and marine boilers. This kind of gauge consists of a movable magnet inside of the boiler, which controls a needle on a dial on the outside, the connection between the two being entirely magnetic.



Thompson Indicator.

THE STEAM-ENGINE INDICATOR.

The steam-engine indicator is an instrument designed to show the pressure of steam in the cylinder at each point of the piston's stroke. It does this in the following

manner: A pencil, moving up and down with the varying pressure of the steam, draws a line on paper, which has a motion backward and forward coincident with that of the piston.

The paper is placed on a drum, which, while the piston is advancing, is caused to make about three-quarters of a revolution by means of a cord connected with a suitable part of the engine, and, while the piston is receding, is brought back to its first position by the reaction of a spring. The pencil is attached to a small piston moving without friction in a cylinder, and the motion of which is resisted by a spring of known elastic force.

The pressure of the atmosphere is always on the upper side of this piston, and when the communication with the cylinder of the engine is closed, it is on the under side also; and if then the motionless pencil be applied to the moving paper, it will draw a line which is called the atmospheric line.

When the communication is opened between the under side of this piston and one end of the cylinder of the engine, the piston will be forced upward by the pressure of the steam, or downward by that of the atmosphere, as the one or the other preponderates. And if now the pencil be applied to the moving paper, it will describe, during one revolution of the engine, a figure, each point in the outline of which will show, by its distance above or below the atmospheric line, the pressure in that end of the cylinder, when the piston was at the corresponding point of its forward or return stroke.

The spring which resists the motion of the indicator-piston is so proportioned in strength that a change of pressure of one pound on the square inch will cause the pencil to move up or down a certain fractional part of an inch.

The diagram thus described shows on inspection the following particulars, viz.: what proportion of the boiler pressure is obtained in the cylinder; how early in the stroke the highest pressure is reached; how well it is maintained; at what point and at what pressure the steam is cut off; whether it is cut off sharply, or in what degree it is wire-drawn; at what point and at what pressure it is released; in a non-condensing engine, whether it is freely discharged, or what proportion of it remains to exert a counter-pressure; in a condensing engine, the amount of the vacuum, and how quickly or how gradually it is obtained; and, in both classes of engines, whether, before the commencement of the stroke, there is any compression of the vapor remaining in the cylinder, and if so, at what point it commences, and to how high a pressure it rises.

From the diagram, the mean pressure exerted during the stroke, to produce and to resist the motion of the piston, may be ascertained, and thus the engineer may come to know accurately the amount of power required to overcome the whole aggregate resistance on the engine.

It must be borne in mind that the indicator shows only the pressure at each point of the stroke; to represent this faithfully is its sole office. It tells nothing about the causes which have determined the form of the figure which it describes. The engineer concludes what these are as the result of a process of reasoning, and this is the point where errors are liable to be committed.

Conclusions which seem obvious sometimes turn out to be wrong, and the ability to form an accurate judgment as to the causes of the peculiarities presented in a diagram, is one of the highest attainments of an engineer.

TECHNICAL TERMS USED IN CONNECTION WITH THE EMPLOYMENT OF THE INDICATOR.

The term Adiabatic literally means no transmission. As applied to an expansion curve, it means that it correctly represents at all points the pressure due both to the volume and the temperature, just as if no transmission of heat to or from it had taken place.

Admission.—This term is applied to the induction of the steam into the cylinder when the valve opens at the commencement of the stroke.

The term Asymptote means a line which approaches nearer and nearer to some curve, but which, though infinitely extended, would never meet it. The clearance and vacuum lines of a diagram are asymptotes of a true expansion curve.

The letter B at the end of a diagram means that that end was taken from the bottom end of the cylinder.

A. B. or Aba. is understood to stand for above atmosphere, and B. A. or Bla. below atmosphere.

The term Compression is a term used to express the distance through which the piston moves in the cylinder after the exhaust has closed. Compression takes place between the piston and the cylinder-head at the end of each stroke; and the distance from the end of the cylinder at which it takes place depends on the amount of lap on the valve.

The term Cushion means the resistance offered on the opposite side of the piston induced by the steam shut up in the cylinder.

Cylinder efficiency.—This term is used to designate the amount of work performed in the cylinder of a steam-engine for a given pressure.

The term Clearance is used to express the extent of the space which exists between the piston, the cylinder-head, and the valve-face at each end of the stroke.

Displacement.—This term is applied to the cubic contents, or the volume of water, steam, or air displaced by the piston during one stroke. It may be found by multiplying the area of the piston in inches by its stroke in inches. The product will be its displacement in cubic inches.

Duty.—This term is understood by engineers to mean the efficiency of steam-engines, or the number of pounds that an engine is capable of raising one foot high per second with an expenditure or consumption of one hundred pounds of coal.

The term Flexure means bending or curving. The point of flexure in a diagram is the point at which the cut-off closes and the expansion curve begins, as shown at *C*, explanatory diagram No. 1, page 245. The point of contrary flexure is the point at which the line changes its direction by curving outwards and afterwards inwards, as shown at *A*, on diagram on page 245.

H. P. cyl. stands for high-pressure cylinder.

H. P. means horse-power, which, when applied to the steam-engine, means 33,000 lbs. raised one foot high; or 150 lbs. raised 220 feet high; or 550 lbs. raised one foot high in one second.

The term Hyperbola means a plane figure which is formed by cutting a portion from a cone by a plane, parallel to its axis or to any plane within the cone, which passes through the cone's vertex. The curve of the hyperbola is such that the difference between the distances of any point in it from two given points is always equal to a given right line.

The term Isothermal means uniform or same tem-

perature. As applied to an expansion curve, it means that such a curve represents correctly the expansion or compression of the steam when the temperature is uniform.

L. P. cyl. means low-pressure cylinder.

The term Ordinates means the vertical lines drawn across diagrams to facilitate the calculation of their power. See diagrams on pp. 245–252.

The term Parallelism is generally employed where two or more straight lines may be extended indefinitely, without any tendency to approach or diverge from one another. See atmospheric and vacuum lines on indicator diagrams.

Release.—This term is understood to mean exhaust. *Residuary* exhaust is that which follows the first release of the terminal pressure. The term *negative exhaust* is sometimes used, though not generally understood in its literal sense. It means compression or cushion, and absolutely amounts to the same thing, as it is merely an early product of the exhaust, for the purpose of retaining a portion of steam in the cylinder as the crank approaches the centre of the stroke.

Rev. or Rev's is understood to mean revolutions per minute, though *rspm* is sometimes used.

I. H. P. means indicated horse-power. It means the number of H. P. of energy shown by the diagram of an engine, as found by multiplying together the area of the piston in square inches, its speed in feet per minute, and the mean effective pressure shown, and dividing the product by 33,000.

N. H. P. means net horse-power, which is the I. H. P. minus the friction of the engine.

The term initial pressure is generally understood to mean the pressure represented in the cylinder between the

opening of the steam-valve and the closing of the cut-off. More properly speaking, it is the pressure represented in the cylinder at the commencement of the stroke, as the pressure frequently falls considerably before the closing of the cut-off.

M. E. P. means mean effective pressure. It is simply the amount by which the average impelling pressure exceeds the average resisting or counter-pressure. The M. E. P. on the piston of a steam-engine is the measure or exponent of the work performed.

The term Terminal pressure means the pressure at which the steam is exhausted from the cylinder, and may be said to be the exponent of the consumption of water by the engine.

The term Pipe diagram is applied to diagrams taken from the steam-pipe for the purpose of determining how much of the pressure of the steam in the pipe is lost in passing through the steam-ports to the cylinder.

The term Scale means the number of pounds of steam per square inch (acting on the piston of an engine) represented by each inch of vertical height on the diagram. Thus a 40 pound scale means that each inch on the diagram represents 40 pounds of steam per square inch, and so on.

The term Spring means the spring which is employed on the piston of the instrument, in order to resist the pressure of the steam and the vacuum. The following **table** will give the limit of pressure in the cylinder to which each spring may be subjected. The length of each spring given in the third column is such that each of them would be extended (when subjected to a perfect vacuum) to a length of $2\frac{7}{16}$ inches, which is the approximate length which would carry the pencil to the lower limit of the range of movement above given.

Scale of Spring.	Limit of Cylinder-Pressure above Atmosphere	Length of Spring.
15 lbs. per in.	25 lbs.	2·192 ins. = nearly $2\frac{1}{5}$ ins.
20 " "	38 "	2·255 " = a little above $2\frac{1}{4}$ "
30 " "	64 "	2·315 " = " " $2\frac{3}{10}$ " or nearer $2\frac{5}{16}$ "
40 " "	90 "	2·345 " = nearly $2\frac{7}{20}$ "
60 " "	143 "	2·376 " = a little over $2\frac{3}{8}$ "
80 " "	195 "	2·391 " = a little above $2\frac{2}{5}$ "

To find the corresponding limit for grades not given, *multiply* the total range of movement, 2·625 inches, by the scale of the spring, and deduct the pressure of the atmosphere.

Example.—Suppose it is desired to find the limit of pressure for a 50 lb. spring: $50 \times 2{\cdot}625 - 14{\cdot}7 = 116{\cdot}55$.

The term String, as used in these pages, means the aggregate length of the ordinates of an indicator diagram.

The letter T on a diagram denotes that that end was taken from the top end of a cylinder.

The term Undulating means rising and falling, wavy.

Wire-drawing.—This term is applied to the common method of regulating the flow of steam from the boiler to the cylinder, by throttling or forcing the steam to ooze through some small or intricate device, such as the governor-valve, thus tending to destroy its elastic force.

The term Zero, when applied to indicator diagrams, means a vacuum.

WHAT INDICATOR DIAGRAMS SHOW, AND HOW THEY SHOW IT.

The object of indicator diagrams is to show the pressure acting on the piston of the engine to which it is ap-

plied at all points, and also at what part of the stroke any change of pressure takes place.

Indicator diagrams supply the means by which to calculate the mean effective pressure acting on the piston, which, together with the known area and speed of the piston, furnishes the factors from which to calculate the power of engines.

Indicator diagrams show the steam-pressure by the height to which the pencil traces the line on the paper measured from the atmospheric or vacuum line.

When the line representing the back-pressure in the diagrams of high-pressure engines shows more than one pound above atmosphere, or, in low-pressure engines, two or three pounds more than the vacuum-gauge shows in the condenser, the diagram indicates undue back-pressure, and that there is evidently something wrong.

The diagram shows whether the valves of a steam-engine are properly set or not, because if there is too little lead it will lean towards the exhaust. If the exhaust takes place too early, the point, *D*, in diagram No. 1, page 245, will be further from the end, *I*; whereas, if the exhaust closes too early, and, as a consequence, there is too much "cushion" or "compression," it will be shown by the great distance of the point *F* from *E*.

A diagram shows whether the piston and valves are leaky or not; though it is often difficult to decide to which the leakage may be due, as the one neutralizes the other. But if the piston alone leaks, the effect will be a more rapid fall of the pressure during expansion than theory requires, and the back-pressure will be greater than if the piston was tight. If the slide-valve leaks, the effect on the diagram will depend on the point at which the leakage occurs. It may leak at the ends, so as to keep on admitting steam after it covers the port; or it may leak

at the bridges, and allow the steam to escape in advance of the exhaust. In the first case, the expansion line would fall less, and in the latter case more, than theory requires.

A diagram shows whether the steam is throttled or not by the expansion curve falling below the boiler-pressure when the throttle-valve is wide open.

A diagram shows the effect of small ports and small steam connections by the steam-line starting below boiler-pressure, and falling before the closing of the cut-off. A pipe-diagram is the only reliable means of determining such defects.

A diagram shows the effect of exhaust-lead, by the exhaust taking place before the end of the stroke is reached, as in nearly all the diagrams shown.

A diagram shows that the indicator is out of order, or whether there is lost motion between the piston and the pencil lever, by indicating more back-pressure than actually exists.

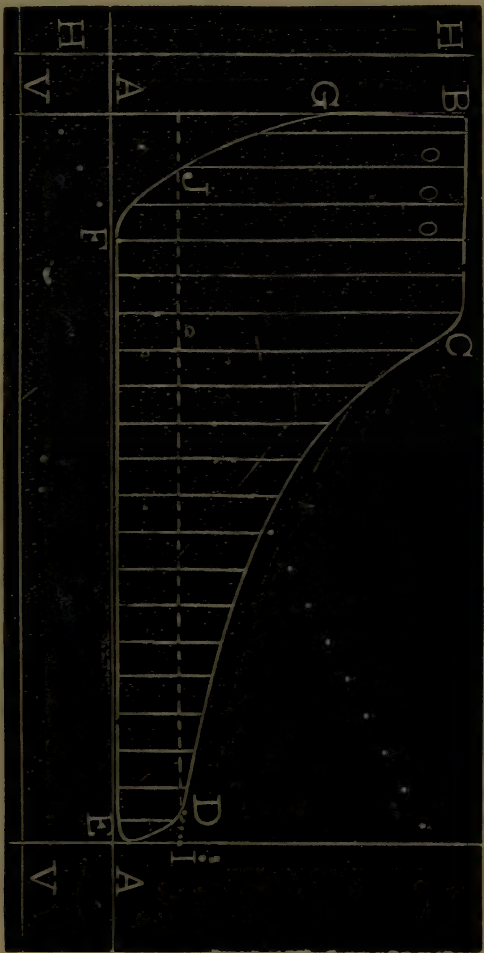
A diagram shows the point of cut-off, which may be termed the point of contrary flexure, that is, the point where the steam-line, *B C* (explanatory diagram, page 245), changes its direction from a straight line to a curve.

A diagram shows the state of the vacuum in the condenser, and whether too much or too little injection-water is used or not; but in this case it is less reliable than the vacuum-gauge. Too much injection-water can only be shown on the diagrams, by taking one first with the proper quantity, and another with the increased quantity, and calculating the power of each. If the extra power, required to pump out the extra water against the atmospheric pressure, more than counterbalanced the gain from the better vacuum, the conclusion would be that too much injection-water was used.

INDICATOR DIAGRAMS.

All indicator diagrams are the perfect pictures of the performances of the engines from which they are taken, provided the indicator is in good order. There are two

Explanatory Diagram No. 1.



senses in which a diagram is said to be perfect or imperfect. First, it may be in perfect conformity to existing conditions, as clearance, load, steam-pressure, etc., though all of these conditions may be far from the best; or, second,

it may not only conform to the above conditions, but it may represent the best attainable conditions, which would include no clearance at all, which is unattainable.

In diagram No. 1, *B C* shows the steam line; *C*, point

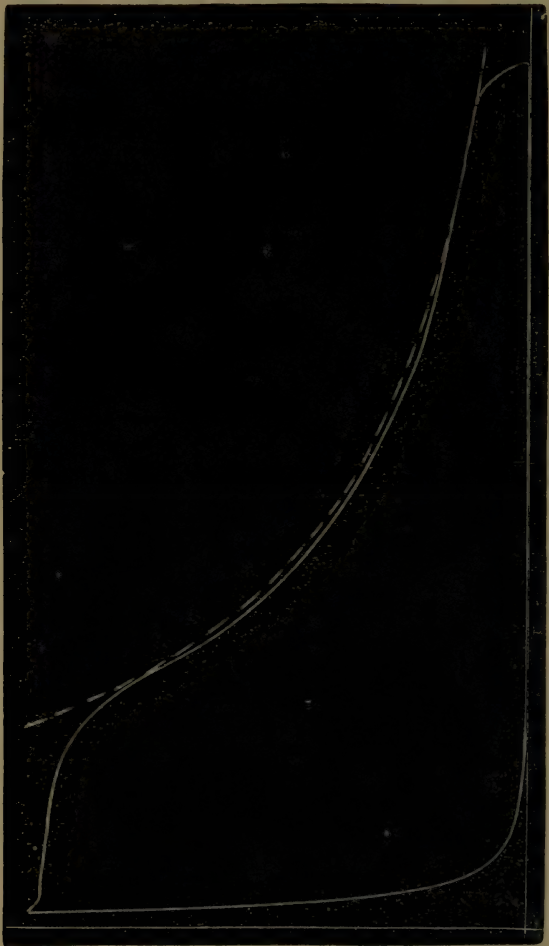


Diagram No. 2.

of cut-off; *C D*, expansion curve; *D*, exhaust; *D E*, exhaust line; *E F*, counter-pressure line; *F*, point of exhaust-closure; *F G*, compression curve; *G B*, admission line; *A A*, atmospheric line; *V V*, vacuum line; *H H*, line representing the clearance; 0 0 0, ordinates for ascertain-

ing the average pressure; *I*, continuation of the expansion curve to end of stroke, to give the terminal pressure for the purpose of calculating theoretical consumption; *J*, the point in the compression curve where the pressure equals

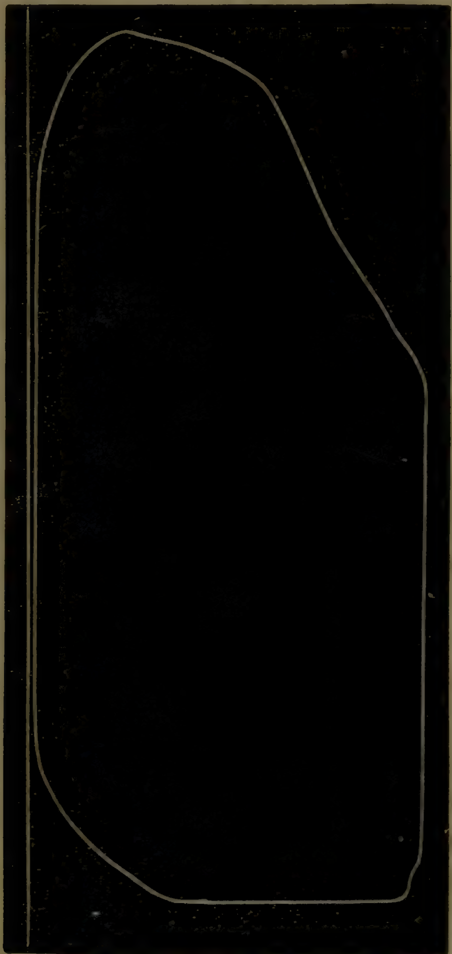


Diagram No. 3.

the terminal; consequently, *I J* is the proportion of the whole stroke taken as the measure of the consumption.

Diagram No. 2 was taken from a Buckeye automatic cut-off engine 22 × 44; piston speed, 520 feet per minute; scale, 40; clearance, 1·75 per cent.; mean effective press-

ure, 36 lbs. It shows very perfect performance both of the engine and indicator.

Diagram No. 3 was taken from a locomotive built at the Baldwin Locomotive Works, for the Pennsylvania Railroad Company, to run on the Philadelphia and Erie Railroad. Diameter of cylinder, 18 inches; stroke, 22 inches; speed, 93 revolutions per minute; boiler-pressure, 115 lbs. per square inch; initial pressure, 100 lbs.; mean effective pressure, 86·60 lbs.; clearance, 4 per cent. At the time the diagram was taken, the engine was pushing a train of 15 loaded cars, whose gross weight was 302 tons, throttle-valve wide open, against a grade of 74 feet rise per mile. Adhesion per ton of load 600, resistance per ton due to grade 35·7 lbs. The slight rounding of the induction corner was probably caused by too much pressure on the pencil, which prevented it from rising till after the paper started to move. The diagram is very good. The expansion curve, as far as can be observed from its limited extent, is correct, and its compression curve very nearly so.

Diagram No. 4 was taken from the high- and low-pressure cylinders of the compound engines of the steamship St. Paul, built by Cramp & Sons, of Philadelphia, on her trial trip, and now plying between San Francisco, Cal., and Alaska. Scale of high-pressure cylinder 30 lbs., of low pressure cylinder 12 lbs., per square inch. The data are as follows: steam, 67 lbs.; revolutions per min., 74; cut-off, ·25; vacuum, 26; indicated horse-power of high-pressure cylinder, 262·5; of low-pressure cylinder, 265·63; total, 528·13. Mean effective pressure of high-pressure cylinder, 43·125; of low-pressure cylinder, 14·25 lbs. The terminal pressures, as shown by the diagram, are as follows: The mean terminal-pressure of both ends of the high-pressure cylinder is 47 lbs. (above vacuum);

volume, 550. Of the low-pressure cylinder is 11·25 lbs. above vacuum; volume, 2100. The equivalents for each cylinder of the combined power of both are as follows:

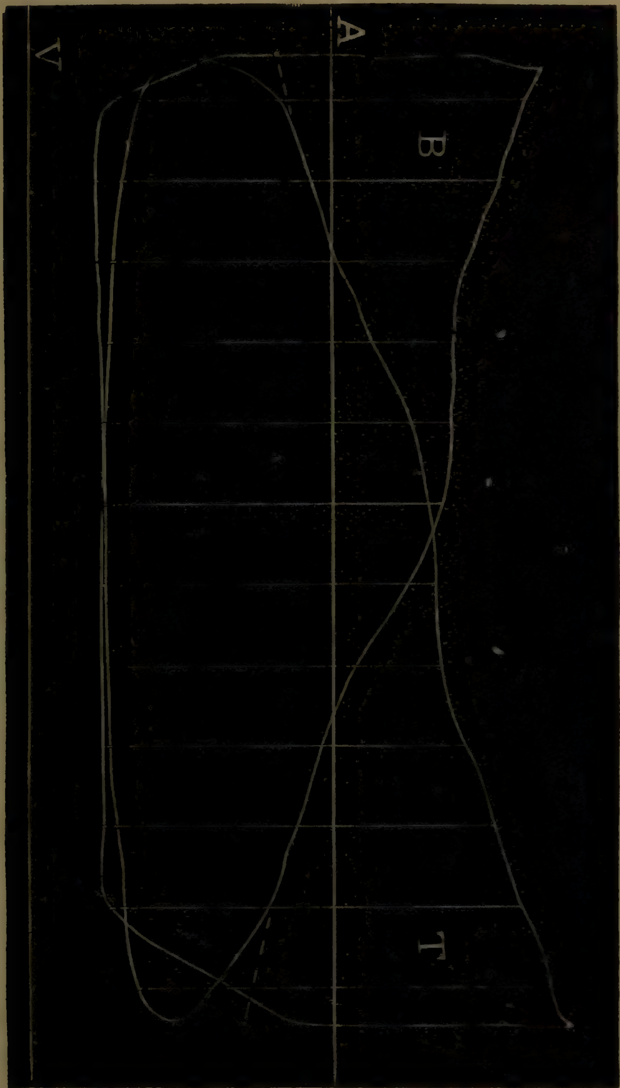


Diagram No. 4.

For the high-pressure cylinder, 43·125 + 43·64 = 86·765.
For the low-pressure cylinder, 14·25 + 14·082 = 28·332.
From these data, the calculation of the theoretical rates

of water consumption will be for each cylinder as follows:

For the high-pressure cylinder, $\frac{859 \cdot 375}{86 \cdot 765 \times 550} = 18$ lbs.

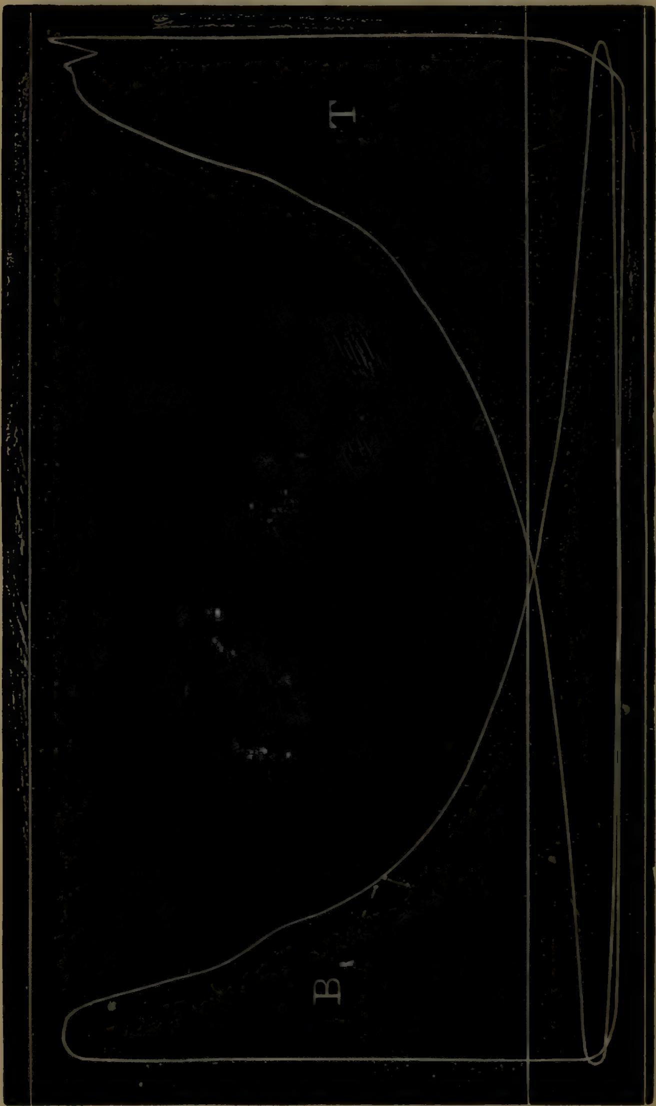


Diagram No. 5.

per indicated horse-power per hour. For the low-pressure

cylinder, $\frac{859{\cdot}375}{28{\cdot}332 \times 2100} = 14{\cdot}44$ lbs. indicated horse-power per hour.

Diagram No. 5 was taken from the simple surface-condensing engine of the steamship Vera Cruz, of Alexander's Line, on her forty-fourth return voyage to New York from Havana. It represents considerably lighter load than diagram No. 21,* and shows the attainment of a better vacuum, is more perfect in its lines, and is equally correct in its expansion curves. The line above the diagrams represents the boiler-pressure. The calculations are as follows: Mean effective pressure of diagram *B*, 17 lbs. Mean effective pressure of diagram *T*, 19·5 lbs. Mean of the two, 18·25 lbs.

Terminal-pressure of bottom diagram, . .	6· lbs.
Terminal-pressure of top diagram, . .	7· lbs.
Mean of the two, ...	6·5 lbs.

Taking 3600 as approximately the volume of 6·5 lbs. pressure, the rate of water consumption will be 13·08 lbs. per *indicated horse-power* per *hour*, which, if equalled, has never been exceeded by any other engines in this country, either simple or compound.

Formulæ for Finding the Horse-Power of Steam-Engines by Indicator Diagrams.

The custom of dividing the indicator card into ten ordinates has been generally adopted by engineers because ten is the most convenient number for a divisor, since the process of dividing by it consists merely of pointing off one decimal. The M. E. P. is ascertained by dividing the aggregate length of the ordinates by their number, and multiplying the quotient by the scale of the diagram.

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* See Roper's "Engineer's Handy-Book" page 311.

The following instructions will be found useful to persons unaccustomed to make the calculation.

First.—*Divide* the card into ten equal parts, as shown by the dotted lines in the diagram below, after which draw a line exactly through the centre of each space, as shown by the full lines 1, 2, 3, etc. Then draw the dotted line *A A*, representing the atmospheric line; also draw the full line *V V*, representing the zero, or vacuum line, which

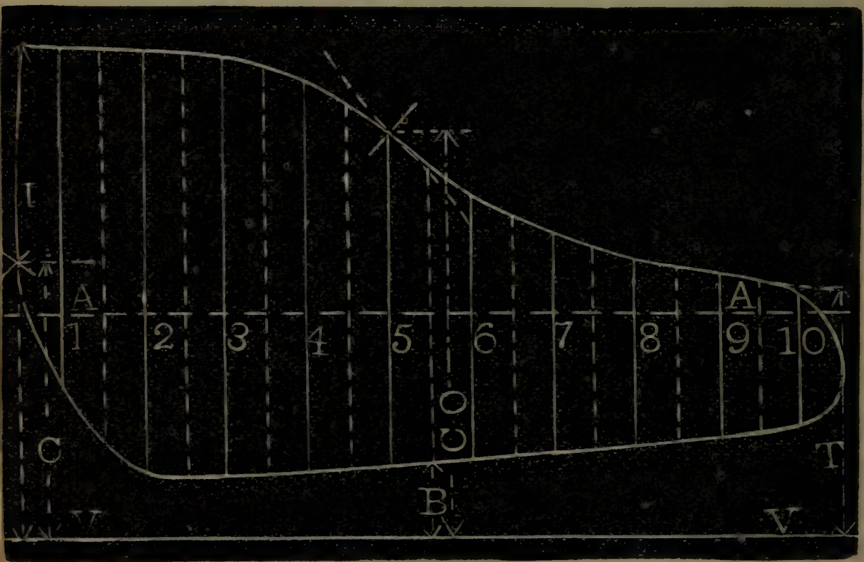


Diagram No. 6.

is equal to $14\frac{7}{10}$ pounds, below the atmospheric line; then measure the card at the following points:

The initial-pressure as shown at . . .	*I.*
The pressure at the point of cut-off . .	*C. O.*
The terminal-pressure at	*T.*
The pressure at the end of the cushion .	*C.*

Next measure the full lines, or ordinates 1, 2, 3, etc., with a slip of paper, marking with a sharp pencil or the point of a knife the length of each, until it contains the

sum of all their lengths, which in this case will be found to be 11·75 inches; then, from the mean length $\frac{11{\cdot}75}{10} =$ 1·175 inches, and the mean-pressure 1·175 × 16 scale of the indicator = 18·80 pounds; the correct rendering of such a card would be as follows:

Initial-pressure	(above zero) =	*I.*	=	32·0 lbs.
Pressure at cut-off	" " =	*C. O.*	=	28·0 "
Terminal-pressure	" " =	*T.*	=	17·0 "
Mean back-pressure	" " =	*B.*	=	5·6 "
Pressure at end of cushion (above zero)	=	*C.*	=	18·5 "
Mean-pressure	=			18·8 "

Suppose the diagram to be taken from one end of a cylinder 50 inches in diameter (with a stroke of 48 inches), making 50 revolutions per minute, and the area of piston to be 1963·5 square inches, then 1963·5 × 18·8 = 36,913·8. This pressure acts on the piston throughout the stroke, 48 inches, 50 times a minute, and the work done on one side of the piston in each minute would be $36{,}913{\cdot}8 \times 50 \times \frac{48}{12}$ = 7,382,760. Now, if another diagram were taken from the other end of the cylinder, and the measurements be the same, the total work done by the engine each minute would be 3000 = 447, indicated horse-power.

HOW TO KEEP THE INDICATOR IN ORDER.

After the indicator has been used, and before putting it away, it should be taken apart and carefully cleaned and dried, to prevent injury to the springs, and to keep the dust and dirt from scratching the cylinder and piston; and, before using it again, the cylinder and piston, and the axis

of the paper cylinder, should be lubricated with some clean oil. If, in the use of the indicator, the cylinder or piston should be cut or scratched, so as to interfere with the freedom of its motion, they should be delicately scraped and burnished, or ground with some nicely prepared polishing powder or tripoli.

THE DYNAMOMETER.

The dynamometer is frequently employed instead of the indicator to measure the power transmitted from the engine to the machinery. In using the dynamometer, it is necessary to have the driving-pulley or band-wheel of the engine loose on the shaft, in order that it may move around freely. It is then connected by the springs of the dynamometer to a clamp, bolted firmly to the shaft, so that the strain on the band-wheel will cause it to turn on the shaft and act on the springs, producing a push or pull; consequently, for every revolution the engine makes, an amount of power will be exerted which will be represented by the strain, in pounds, on the springs connecting the driving-pulley to the clamp on the shaft.

This strain multiplied by the number of feet through which the mechanism passes, and this number of foot-pounds multiplied by the revolutions per minute and divided by 33,000, will show the amount of work, in horse-power, that the engine is performing.

To ascertain the power exerted by the engine of a screw-vessel, the thrust of the screw is made to bear upon the fulcrum of a lever of the second class, by receiving the force near the fulcrum; and having a long arm for the weight, the force exerted by the screw is thus decreased in a great and easily ascertained ratio, somewhat after the manner by which, in the weighing-machine, a small weight in the machine-house balances a considerable one on the platform.

Machinery very rarely transmits power uniformly from one locality to another; which is particularly the case with the ordinary steam-engine, as the storage and delivery of work by a fly-wheel causes an irregularity in the power transmitted which can be measured by the dynamometer. But the dynamometer is best adapted for measuring the force of pulling a load on a road, a boat on a canal, or of towing a ship. The force in pounds indicated by the dynamometer, multiplied by the velocity in feet per second, will be the power in effect, which divided by 550 will give the horse-power in operation.

The force driving a paddle-wheel is frequently measured by a dynamometer placed on shore, a rope being carried from the vessel and fastened to the dynamometer, when the engines are set to work, and their tractive force ascertained precisely as in the last case. The use of the dynamometer has greatly furthered the mechanical improvement of screw-engines by affording facilities to estimate the thrust of the screw, and thus ascertain if any large amount of force is being wasted.

Rule *for finding the Dynamometrical or Effective Horse-power of a Marine Engine.*—Having found (by the dynamometer) the number of pounds pressure exerted by the screw-shaft, multiply it by the speed of the ship in knots, and the product by 6080 (the number of feet in a knot); then divide the result by 60 (the number of minutes in an hour) and by 33,000, and the quotient will be the horse-power.

Another Rule.—Multiply the number of pounds pressure by the speed of the ship in knots, as before, and this product by ·00307; the product will be the horse-power.

THE ENGINEER.

The skilful and practical engineer is a very important man, either in a manufacturing establishment, on a locomotive, or on a steamship. And it can be safely said, that there are as many instances of genuine worth and ability to be found among engineers as in any other trade or profession — men who from small beginnings have worked themselves up to important positions, and have, by their intelligence and capability, not only won the respect of their employers, but of all with whom they have come in contact.

Unfortunately, this is not so with all, as engineers are frequently met with who claim to know everything mentioned to them, and that they knew all about it long ago, or that they had a hand in originating the idea themselves. There is also a great tendency among persons having charge of steam-machinery, to look upon steam as something very mysterious; and this tendency is not always confined to engineers who have the immediate charge of steam-engines and steam-boilers, but among those who, from ability or some other influence, occupy higher positions. Now an engineer should be sure that his views are correct before putting them forth, and he should also be modest in expressing them, especially in the presence of his superiors. He may be able to teach others in his profession, but in communicating his views, he should avoid making them feel that he assumes any superior knowledge. Men of superior ability generally prefer to show by their work what they know, and if a reason is asked for this or that, they are always ready with a clear and concise answer.

The opportunities in this country for young engineers to rise are equalled in no other country in the world; for this reason they should improve every opportunity to

qualify themselves for the responsible duties of their calling, as it is only by slow and careful study that they can arrive at the logical conclusions so essential to success in their profession. They must remember that it is not through inspiration that the expert attains more accurate results than the novice; yet perhaps the former was once more awkward and rude in science than the latter, and only obtained his superiority and skill by close study and investigation. It frequently occurs, however, that the expert in engineering, as in all other professions, is only an expert in name.

For this reason there is a strong prejudice in favor of those who are known to be practical men, in distinction from those who are called theoretical engineers. The practical engineer is understood to be one who relies entirely upon the information he has gained by his personal experience, while the so-called theoretical engineer is willing to accept the facts established by others, when they are well authenticated, and uses them to increase his knowledge. It is hardly necessary to state that of two men, each having the same natural intelligence, the one who employs his intellect, and adds the result of his studies to the knowledge that he has gained by experience, will in general be much the abler engineer of the two. It is true, that if he relies entirely upon theory untested by experiment, his views will be of little value, since that theory only is correct which takes account of all the conditions that occur in practice.

The very nature of sea service calls for superior intelligence in those on whom depend the care and management of a ship's machinery, on account of the very serious nature of the results which may accrue from a failure of the power in an emergency. Engineers should, therefore, prepare themselves for any casualty that may arise, by consider-

ing possible cases of derangement, and deciding in what way they would act should certain accidents occur. The course to be pursued must have reference to particular engines, and no general rules can therefore be given; but every marine engineer should decide on certain measures to be pursued in the emergencies in which he may be called upon to act, and where everything may depend upon his energy and decision.

When engineers, or any other class of mechanics, fail to improve or qualify themselves to discharge the duties of their respective callings with ability and honor to themselves, their trade is sure to degenerate, until, from being a profession or a science, it falls into the hands of incompetents, and ceases to be anything more than a mere occupation. Ignorance among any class of mechanics is a great misfortune, but it is particularly so in the case of engineers.

MANAGEMENT OF LAND AND MARINE ENGINES.

Extensive as is the literature connected with the steam-engine, there is very little in print in relation to the practical management of steam machinery. It is not difficult to discover the reason for this omission. The practical details are so varied, for the different cases that may arise, that it is almost impossible to classify them.

Among the most important duties of a marine engineer are, the proper adjustment of the different parts of the machinery, and to see that they are neither too tight nor too slack, and that none of them becomes injured from heating. In the generality of marine engines, the bearing most apt to heat is the crank-pin; but much depends on the proportions of the parts, which differ in different engines. But, as in most engines the crank-pin can be touched with the hand at each revolution, there can be but little excuse for allowing any serious heating at that point.

In cases of extreme heating of the crank-pin, it is usual to slack up on the keys, and lubricate the parts in contact with a mixture of sulphur and oil, or tallow, lead filings, or quicksilver; but it sometimes becomes necessary to cool the heated parts with cold water applied by means of a hose communicating with the deck-pumps. Such extreme heating rarely occurs, except when the keys are too tightly driven, or the regular supply of oil neglected.

When pillow-blocks, or main bearings, become troublesome from heating, the annoyance, in a majority of cases, can be remedied by mixing a quantity of Bath brick-dust with water, and running it through the holes in the caps when the engine is in motion, as it has a tendency to smooth off the surfaces in contact, and bring them to a solid bearing. In cases where heating of the heavy revolving parts is induced by grit, or such foreign substances as are frequently found in inferior qualities of oil, the difficulty may be removed by using a strong solution of potash, or concentrated lye, on the parts affected while they are in motion. They should be thoroughly lubricated immediately after the solution is applied.

Looseness in any of the revolving parts generally manifests itself by a knock; but the keys of the parts that only vibrate may drop out and cause serious damage, without giving any warning; for this reason, when the keys are properly adjusted, the set-screws should be screwed up sufficiently tight to prevent the possibility of the key moving backward or forward. Generally speaking, keys have a tendency to work further in, which frequently causes serious heating before the engineer is aware of it.

While the vessel is in port, the bonnets and casings of the steam-chest should be removed for the purpose of examining the valves, faces, and seats, and if any hard or cut places be found they should be carefully scraped and

refitted. The crank should then be placed on the top and bottom centres, with the go-ahead gear in position, in order to see whether the valves open and close at the right time, or if they have the proper amount of lead.

The piston should frequently be removed from the cylinder, and the faces of the rings, where they form the joints with the flange of the piston-head and the follower-plate, reground and fitted, and the spring packing readjusted. The tightness of the piston should also be proved, by admitting steam above or below it, and opening the indicator's cocks on either side, to see if any steam escapes; in such cases, the injector-cocks should be slightly opened an instant, to withdraw any steam that may have collected on the opposite side of the piston, so that the passage of any steam may be more readily perceived. The tightness of most parts of the engine may be tested in this way without moving it more than half a stroke.

The link-motion should next receive special attention, for the purpose of ascertaining if the link, link-block, link-pins, eccentric-straps or rods, need readjustment or repairs. The screw-shaft should also be carefully examined, to determine if lining-up is required, or if the gland or any part has become badly worn or seriously cut.

The air-pump cover should then be lifted, and the bucket withdrawn, for the purpose of ascertaining if the foot-valves are in good order. The condenser should also be proved, which can be done by taking off the door or doors, and filling it with cold water; and should any leak be discovered, the tube or tubes should be removed and repaired or replaced with new ones. Every engineer should make himself perfectly familiar and conversant with all the details of the surface condenser.

The state of the vacuum will be shown by the vacuum-gauge attached to the condenser; and if it be imperfect,

the cause must be ascertained and the fault corrected. If the hot-well be much more than blood-warm, more injection water must be admitted; and if the vacuum is still imperfect, there must be some air leak, which the engineer must endeavor to discover. Very often the fault will be found to lie in the valve or cylinder cover, which must then be screwed down more firmly, or in the faucet-joint of the eduction-pipe, the gland of which will require to be tightened, or the leaking part puttied up. The cylinder and valve stuffing-boxes may at the same time be supplied afresh with tallow, and the door of the condenser examined. The joints of the parts communicating with the condenser are usually tried with a candle, the vacuum sucking in the flame if the joint be faulty.

When a leakage of air into the condenser, or its connections, has been discovered, it may be stopped temporarily by calking in spun-yarn, or driving in thin fine wedges; if the leakage be into the condenser, it is sometimes convenient to allow water to be injected through the orifice, by which means little harm is done. In several cases where, during a long voyage, the bottom of the condenser has become leaky by corrosion, (often induced by galvanic action with the copper bolts of the ship's bottom, as well as the brass foot-valve, etc.,) a water-tight crank has been constructed at sea between the side keelsons. By this means, the condenser and air-pump are submerged in a kind of well constantly replenished with cold water from the sea, which, forcing its way through the leaks by the pressure of the atmosphere, shares with the proper injection water the duty of condensing the steam — the injection-cock orifice being partially closed in proportion to the extent of the leakage through the bottom.

When the vessel is laboring in a heavy sea, the supply of injection water should be diminished; for in such cases,

where the speed of the engines is subject to great and constant fluctuations, depending upon the greater or less submersion of the wheels or screw-propeller, the condenser is liable to become choked with water, thereby causing the engines to stop. The effect of working the engines with a stinted supply of condensing water is, of course, that the condensers will become hot, and the vacuum will be diminished; but this is a minor evil in comparison with endangering the machinery by subjecting it to too severe a strain.

Care must be taken, when the engines make a temporary stoppage, that the injection-cock or air-pump does not leak, and allow the condenser to fill with water, which causes much trouble and delay in starting the engines again; so, should this be apprehended, the sea-cock must also be closed at the same time with the injection-cock.

When a stationary engine is stopped, even for a short time, the cylinder drip-cocks should be immediately opened, in order to allow the water of condensation to escape. They should not be closed until after the engine has been started. Before starting any engine, if it has been standing still for some time, the cylinder should be warmed by admitting steam, and working the engine back and forth with the starting-bar. This is a necessary precaution against the dangers arising from an accumulation of water in the cylinder induced by the steam coming in contact with the cold iron.

The oil or tallow intended to lubricate the cylinder and valves should not be admitted until after the engine has been in motion and the drip-cocks closed; as, otherwise, instead of being returned with the exhaust, and lubricating the rubbing surfaces, a portion of it would be driven out with the water of condensation, and lost.

In setting up, repairing, or driving the keys of steam-engines, a soft hammer or piece of hard wood should

invariably be used to drive the parts fast together. But, in the absence of either, a piece of sheet copper or brass should be interposed between the face of the hammer and the part to be driven. Any engineer can make himself a soft hammer by cutting a hole, for the handle, through a piece of brass or copper tube about two inches in diameter and four or five inches long, and, after inserting the handle, filling the tube with Babbit-metal or lead.

Raising Steam and Getting under Way.—The first duty of the engineer, preparatory to getting under way, is to fill the boilers with water to the upper gauge-cock. If they be located in the hold, it will be only necessary to open the blow-cock, and the water will flow into the boilers through the bottom of the vessel, otherwise it will be necessary to fill them by the hand force-pump; although donkey-pumps having separate boilers are most frequently used for that purpose.

The next step is to start the fires, which should be allowed to burn slowly, in order that all parts of the boiler may expand uniformly, and the safety-valve be kept open for the purpose of allowing the air to escape from the boilers. As soon, however, as steam begins to escape through the safety-valves, they should be immediately closed, for then the air is all expelled, an atmosphere of steam having taken its place.

When the steam-gauge shows any excess of pressure over the atmosphere, the valves may be raised, and steam allowed to flow into the cylinder and through all the pipes; this expels the air and warms the cylinder, and prevents the condensation of steam when the engine is started.

When sufficient steam is shown by the gauge to work the air-pump and produce a vacuum, say 5 or 6 pounds, the injection-cocks should be opened a little, and the eccentric-hook unshipped, and the valves moved back and

forth with the starting-bar or the link, as the case may be, in order to produce a reciprocating motion in the piston. The engine should then be "turned over" two or three times for the purpose of seeing if everything is all right. If everything is found to be in perfect order, the engine is stopped, the oil-cups filled, and all the rubbing and revolving surfaces thoroughly lubricated; then the vessel will be ready to proceed on her voyage.

When it becomes necessary to stop the engine, the steam is first shut off, or nearly so; the supply of injection water diminished, the eccentric-catch unhooked, and the valves worked by hand; the damper in the chimney should also be closed, and the furnace-doors opened.

To back an engine, where only one eccentric is used, the steam is first shut off, the eccentric-hook thrown out of gear, and steam admitted to the opposite end of the cylinder by means of the starting-bar. If the link be employed, it is only necessary to shift it to the backward motion.

Every ocean steamer should carry a liberal supply of duplicates of the parts that would be most likely, in case of breakage, to disable the engine. They should also have a good supply of bolts, nuts, and washers, packing-solder, charcoal, portable forges, hammers, wrenches, spanners, screw- and monkey-jacks, ratchets and ratchet-drills, cold chisels, key-sets, files, reamers, pinch-bars, straight edges, T squares, brass-sheaved blocks, and all such tools as would be likely to be called into play in any emergency, which should be hung up or stored in conspicuous, convenient, and accessible places, for, unless this be done, they are liable to become mislaid or eaten up with rust, as neglect generally follows their stowage in unfrequented or obscure places.*

* See page vi.

HOW TO PUT THE ENGINES IN A STEAM-BOAT OR SHIP.

The art of placing engines in ships is more a piece of plain common sense than any other feat in engineering; consequently, every engineer that engages in such an undertaking must settle a mode of procedure for himself, as it would be impossible to give any general instructions for such work that would meet all the requirements of the varying circumstances of each individual case. But as the subject is one of great interest and importance to engineers, it may not be out of place to offer some general observations upon it, together with specific directions in such particulars as seem to require them; the most practical mode of procedure being as follows:

The first business of the engineer is to ascertain the precise beam centre of the boat. He then erects perpendicular straight-edges towards each end of the boat, sufficiently far apart to clear the cylinder at one end and the shaft and crank at the other. These straight-edges must rise strictly perpendicular to the side level of the boat, because they are to serve as a guide in establishing the sidewise centre lines of the cylinder, gallows frame, walking-beam, and main connecting-rod; and, indeed, must be kept in view in the whole operation of placing the engine in the boat.

The manner of placing a straight-edge in its true position is, to rest the lower end upon the centre keelson, in such position that one side of the piece forming the straight-edge shall be exactly in the beam centre of the boat. To carry up the straight-edge in strict perpendicular from this centre, a straight-edge must be laid also across the boat, at the level of the deck, resting in an exact horizontal position by means of blocks placed under each end, at the sides of the boat. The exact position of

the perpendicular straight-edge may now be ascertained, either by means of a T square or by measuring from the outside of the hull towards the centre.

Having found the position of one perpendicular straight-edge by the means described, the second will of course be fixed in an exact relative position to the first. The proper height of the gallows frame, where the pillow-block rests upon it, must now be measured from the flooring of the boat upon which the engine keelsons rest. This height must always be specified in the working drawings of the engine, to which reference must be had; and having been measured on either side, that side of the gallows frame must be cut off at the proper point to receive the beam pillow-block. A T square applied to the side that has thus been cut off will indicate the point of cutting off the other side of the gallows frame, bringing both sides to exactly the same height. In applying the T square for this purpose, care should be taken to keep the long or perpendicular arm of the square in exact line with the perpendicular straight-edges above mentioned.

The beam pillow-blocks can now be placed in their positions, precaution being taken to see that these blocks are of equal dimensions from the point resting upon the gallows frame to the centre of the journal, any difference to be obviated by the variation of the height of either side of the gallows frame from the exact point heretofore reached. A beam main centre piece of wood must next be made of the same dimensions as the beam centre itself. This wooden beam main centre piece must be placed in the journals of the beam pillow-blocks. The middle of this centre piece, measured from each journal, and indicating the beam centre of the working-beam, must be marked upon it, either by the person turning it or by the engineer himself, usually by the former.

A piece of small cord, of very perfect manufacture and very strong (catgut is generally used), must now be employed, stretched from one straight-edge to the other. A short straight-edge may also be fastened, with screws, to the wooden centre piece, exactly at the middle thereof, indicating the beam centre of the working-beam. This centre line must correspond with the catgut line drawn from the two straight-edges. The wooden centre piece should be brought to rest in exact right angles to the several centre lines of the working-beam; that is, in right angle to the horizontal, perpendicular, and beam centres of the working-beam, as heretofore ascertained and described.

Having described the manner of ascertaining the various points to be considered in fixing the beam pillow-blocks in their places, the work becomes merely mechanical, and will be accomplished by such means as suggest themselves to the engineer; which done, the beam pillow-blocks may be permanently bolted to the top of the gallows frame.

The laying of the bed-plate is the next object to receive the attention of the engineer. The bed-plate is laid on two oak planks, which may easily be adjusted to accommodate the variations in the bed-plate. The planking which lies on the engine keelson, and comes in immediate contact with the bed-plate, should be adjusted as nearly as possible to its proper position. This may be done with the use of a **T** square. The exact centre of the steam-cylinder must now be taken into consideration, and, keeping that point in view, the bed-plate can now be placed upon the planking prepared, as above referred to, for its reception.

The centre points where the condenser and air-pump rest upon the bed-plate must now be accurately ascertained, and again the **T** square must be employed to lay the bed-plate true in every direction. This being accomplished, a line must be stretched from the two upright straight-edges before described, and the air-pump and con-

denser centres must be in accordance with this line. Now fix the true perpendicular centre line of the gallows frame, and measure from that line to the centre line of the cylinder, as established in the working drawing. This will settle the exact position of the bed-plate.

The relative positions of the beam centre, as indicated by the cord drawn from the two upright straight-edges, and of transverse centres of the cylinder, condenser, and air-pump, must be indicated by marks with a chisel upon the ends and sides of the bed-plate flanges, preparatory to the removal of the cord when the condenser is placed in position. Before doing this, however, four marks on the upper and lower flanges of the condenser must be made with a chisel, at right angles to each other, corresponding with the centre of the condenser. This centre is ascertained by means of a wooden cross placed in the condenser, the arms of the cross fitting closely to its inside diameter. The same marks, by the same process, must also be made on the upper and lower flanges of the steam-cylinder.

The condenser may now be fitted to its place, care being taken to bring the centre at the top in exact position, measuring from the centre line of the gallows frame as before, and in accordance with the line drawn from the two straight-edges; in other words, that the centre line of the condenser is in exact perpendicular. The work of fitting the condenser to the bed-plate must, of course, be performed upon the "chipping-strips" in the lower flange of the condenser; and when perfected, the condenser may be permanently bolted to its place.

The cylinder bottom is always bolted to the cylinder, and when thus joined, the two are placed together upon the condenser. The marks upon the outside of the flanges will assist in bringing the lower part of the cylinder to its exact centre point. The cylinder must now be fitted to its

place, care being taken, as in the case of the condenser, to maintain the perpendicular of its centre, the same rules governing both cases.

The slides for the cross-head must next be fitted to their places. To this end, a wooden cross must be placed in the lower extremity of the cylinder, the arms fitting closely to its inside diameter. A temporary platform, near the top of the gallows frame, must now be employed, to which a cord should be stretched from the centre point of the cylinder marked on the wooden cross in the cylinder. This cord should indicate a continuation of the true perpendicular centre line of the cylinder, for the purpose of fixing the true position of the slides. In fixing the position of the slides, a wooden piece may be employed to represent the cross-head, with a point in the centre to show where the continuation of the centre line of the cylinder should pass; and when the slides are accurately set, they should be bolted to the flanges of the cylinder.

A brace must now be fitted to the upper flanges of the slides, to retain them in their proper position. And braces should be extended from the slides to the gallows frame. There should also be a diagonal cross-brace connecting the flanges of the slides with the other side of the gallows frame. Four more braces, two on each side, must be extended directly from the slides to the gallows frame. All these braces must be bolted to the flanges of the slides and to the gallows frame.

The piston, with piston-rod and the cylinder cover, may now be put in their places, and then the cross-head put in its place and fastened to the piston. The working-beam, also, may be laid in the beam pillow-block journals.

The setting of the main shaft and out-port pillow-blocks is the next work in order. A cord must be extended, indicating the centres of these pillow-blocks.

These centres are to be ascertained by measuring the height, on the working drawings, from the top of the flooring of the hull to the centre of the shaft, then drawing a horizontal line with a cord from the straight-edge to the perpendicular centre of the gallows frame at the height of the main shaft. A line must now be drawn perpendicularly through the centre of the shaft, parallel, in every direction, with the centre line of the cylinder, when a T square can be employed in determining the centre line of the main shaft, to be indicated by a cord drawn from one out-port pillow-block to the other; the actual height, as well as the distance from the centre of the gallows frame, having been, as previously stated, ascertained by measuring the height, in the working drawing, from the flooring of the boat, and in using the T square, to see that this line is in strict right angle to the cord drawn from the straight edge to the centre of the gallows frame, and also in right angle to the perpendicular line parallel to the centre of the cylinder.

The placing of the main and out-port pillow-blocks must now be proceeded with. The engineer must measure the distance from the centre of the journals to the lower edges of the pillow-blocks, to ascertain the exact height of the resting-places of them above the flooring of the boat, when he will cut the timbers accordingly, with a view to the exact height of the centre of the shaft, fitting these timbers to the lower sides of the pillow-blocks.

The centres of the journals of the out-port pillow-blocks must always be slightly higher than the centre of the journal of the main pillow-blocks, on account of the great weight of the paddle-wheels, and the fact that the sides of the boat will yield more than the centre to the weight of the engine. If the two parts of the shaft, as usually employed in river-boats, lie perfectly true, the

cranks will show no variation in their distances from each other at any point in their revolution.

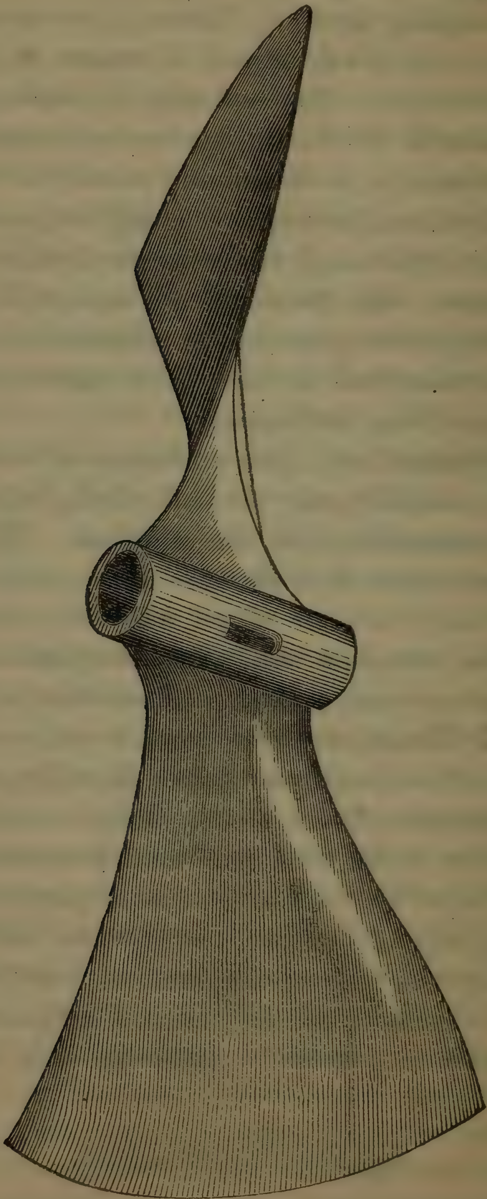
In placing the air-pump in its seat, reference must be had to the working drawing, following the centre points, etc., as there laid down, subject, of course, to the formation of the bed-plate. Mechanically, the operation is precisely the same as in placing the condenser and cylinder. It is well to fasten a piece forming a straight-edge along the engine keelson, the upper or straight-edge of this piece to be in strict right angle to the perpendicular centre line of the cylinder; with the aid of a T square, this straight-edge will supply the place of the perpendicular straight-edges before described, in case their removal should become necessary from any cause.

The bed-plate is laid upon a mixture of white and red lead spread carefully over the oak planks which come in immediate connection with the bed-plate. The object of this mixture of paint is to fill up all the crevices or imperfections of any character which may exist, either in the surface of the bed-plate itself or in the planking, causing the bed-plate to receive equal and substantial support in all its parts. The same process must be followed in laying the main and beam pillow-blocks; these important parts of the engine frequently having been broken in consequence of not being set firmly and accurately in their places. A layer of red lead is also employed to secure a perfectly tight joint between the condenser and bed-plate, after which a rust cement, composed of cast-iron turnings, pulverized sal-ammoniac, and flour of sulphur is calked into the joint to render it perfectly air-, steam-, and water-tight.

SCREW-PROPELLERS.

The screw-propeller, so commonly applied to the propulsion of vessels, consists of two, three, or four helical

or twisted blades, set upon a shaft or axis, revolving beneath the water at the stern. The shaft where it protrudes through the stern of the vessel is surrounded by a stuffing-



Screw-Propeller.

box, containing hemp packing, whereby the entrance of the water into the vessel is prevented, and the extremity of the shaft in the rear of the screw is supported in a socket or bearing attached to the rudder-post. This part rests upon the keel, and from it the rudder is suspended.

The screw revolves in that thin part of the stern of the ship which is called the dead wood, in which a hole of suitable dimensions is cut for its reception; and the thrust or forward pressure caused by the action of the screw upon the water is transmitted to some point within the vessel which can be amply lubricated. It is the thrust of the shaft which is operative in propelling the vessel, and the amount of this thrust can be measured by means of a dynamometer applied to the end of the shaft within the vessel.

The diameter of the screw is the diameter of the circle described by the arms; and the length of the screw is the length which the arms occupy upon the revolving shaft. If a string be wound spirally upon a cylinder, it will form a screw of one thread; if two strings be wound upon a cylinder with equal spaces between them, they will form a screw of two threads; three strings similarly wound will form a screw of three threads, and so of any other number.

If, instead of strings, flat blades be wound edgewise around the cylinder, and each blade has one of its edges attached to the cylinder by welding, soldering, or otherwise, then, if a slice be cut off the end of the cylinder, there will be only one piece of blade attached to that slice, if the screw be of one thread; two pieces of blade, if the screw be of two threads; three pieces of blade, if the screw be of three threads, and so of any number. The number of blades, therefore, of any screw determines the number of threads of which it is composed, and this indication equally holds however thin the slice cut off the end of the screw may be.

The pitch of a screw is the distance measured in the direction of the axis between any one thread and the same thread at the point where it completes its next convolution. Thus, a spiral staircase is a single-threaded screw, and the pitch of such a screw is the vertical distance from any one step to the step immediately overhead.

Ordinary screw-propellers are not made nearly so long as what answers to a whole convolution; and in speaking of their pitch, therefore, it is necessary to imagine the screw to be continued through a whole convolution at the same angle of inclination with which it was begun. Of this whole convolution any given proportion may be employed as a propeller, and the length of a screw, therefore, cannot be determined from the pitch, neither can the pitch be determined from the length.

The form of screw most frequently employed in this country is a screw of two blades or threads, sometimes three or four blades are used. The pitch of the screw is not made less than its diameter, sometimes nearly twice the diameter, and in some instances over twice the diameter of the screw. The length of the screw is usually made equal to one-sixth of the pitch. The thrusting of the screw is measured by the area of the circle described by the arms, which is termed the area of the screw-disk. The screw-disk has generally about one square foot of area for every 2½ or 3 square feet in the immersed transverse section of the vessel.

NEGATIVE SLIP OF THE SCREW-PROPELLER.

By "slip" is meant the difference between the actual advance of the propeller through the water and the advance which would be accomplished, if there were no recession of the water produced by the pressure of the propelling surface.

A screw of 10 feet, if working in a stationary nut, would advance 10 feet for every revolution it performed; but when such a screw acts in the water, it may only advance 9 feet or less for every revolution — the water being, during the same time, pressed back one foot, from its inertia being inadequate to resist the moving force. In such a case, the slip is said to be 1 foot in 10, or 10 per cent.

With every kind of propeller which acts upon water, there must be a certain amount of slip, for any force, however small, will overcome the inertia of the water to a certain extent; but, by so proportioning the propelling apparatus that it will lay hold of a large quantity of water, the backward motion of the water will be small relatively with the forward motion of the vessel; or, in other words, the slip will be reduced to an inconsiderable amount.

One of the most remarkable phenomena connected with the action of the screw is, that under some circumstances its apparent progress through the water is not only as great as that of the ship, but greater. In some of the early experiments with the screw, when the vessel was proceeding under the joint action of steam and sails, it was found that the progress made by the vessel through the water was greater than if the screw worked in a solid nut. It was from this inferred that the ship must be overrunning the screw; yet it was plain that this could not be the case, as the engine was travelling at the usual speed; but investigation of the subject explains the phenomena, and ascribes it to the fact of the screw working in a column of water which follows the ship, instead of in the stationary water of the sea.

When a strong current of water runs through the arches of a bridge, the water may be observed to curl

around those ends of the piers which stand lowest in the stream; and if a chip of wood be thrown into that spot, it will not be carried off by the stream, but will remain at rest, showing that the water is not in motion in that place.

Now, suppose a screw to be placed in this stationary water, it will be obvious that any movement of rotation given to it will produce some thrust upon the screw-shaft; whereas, if the screw were placed in the stream, it would require to revolve faster than the stream runs, before any thrust upon the screw-shaft could be produced.

Now, suppose the pier to be a ship, the other circumstances specified will not be altered thereby; and it is conceivable, that a screw acting in this dead water might aid the vessel to stem the current, even though the screw moved with less velocity than that of the current itself.

That the screw will exert some reaching force upon this dead water, even with any speed of rotation, is obvious enough; but whether, with a speed inferior to that of the stream, it will produce a sufficient thrust to enable the vessel to stem the current, will depend very much upon the shape of the vessel and the dimensions of the screw employed.

If the pitch be fine, and the number of revolutions answering to a given speed of vessel be great, there will be a tendency to pile up the water at the stern, owing to the adhesion of the water to the rapidly revolving blades, and the consequent acquisition of a considerable centrifugal force by the water. When this action occurs, the vessel will be forced forward, to some extent, by the hydrostatic pressure produced by the elevation of the water at the stern, and this pressure will aid the thrust of the screw. If, then, by such an arrangement, a vessel could be made to stem a current, she could obviously, under like conditions, be made to move through still water.

All vessels carry a current in their wake which answers to the dead water in the case of the bridge; and if a screw acts in this current, then the apparent slip will be positive or negative, just as the real slip, or the velocity of the current, may preponderate. In every case, the screw must have some slip relatively with the water in which it acts; but if that water has itself a forward motion, the result cannot be the same as if the water were stationary, and it will be necessary to reckon the forward motion of the current as well as the forward motion of the slip.

Thus, if the real slip of the screw be three miles an hour, and the following current runs at the rate of three miles an hour after the ship, then there will appear to be no slip, if the comparison be made with the open ocean on each side of the vessel; or there will appear to be a negative slip, as it is termed, of one mile an hour, if the following current runs at the rate of four miles an hour. The whole perplexity vanishes, if we consider that a current follows the ship at a rate which may be greater or less than the slip of the screw. This current is confined to the water very close to the ship, so that a log, whether of the ordinary or the patent kind, will not take cognizance of it if thrown over the stern.

The centrifugal action of the screw, it appears not improbable, besides piling up the water at the stern, and thus forcing the vessel on with a velocity which may be greater than that of the screw, also causes a current of water to flow radially from the centre of the screw to its circumference; and this stream of water, by intervening between the surface of the screw and the nut of water in which it works, may assist in making the vessel travel faster than the screw itself.

In all screw-vessels, the slip is greater than is generally supposed; for in all of them there is a following current

in which the screw works; and as, in some cases, the current conspires to make the apparent slip to disappear altogether, so it will, in every case, reduce the visible slip to a less amount than the real slip, and it is the real slip which it concerns us to determine. There is no benefit derived from the existence of a following current in screw vessels, for to produce the current requires a large expenditure of power; and in screws so proportioned as to produce a negative slip, a poorer performance has been obtained than in cases in which screws producing an apparent slip of 10 to 20 per cent. have been employed.

The Screw as compared with the Paddle.—Under favorable circumstances, there is but little difference between screws and paddles. In running before the wind the paddle has the advantage; but when the wind is ahead, it is not so, for the wind acts on the paddle-boxes, which offer great resistance and retard the ship. The superiority of the screw is shown in long voyages; for, whereas the lightening of the ship proves detrimental to the paddle, it cannot be so to the screw, the screw being more deeply immersed. The screw requires deeper water than the paddle. In ships of war, the screw gives a clear broadside, while the paddles occupy room that should be devoted to the guns. The vibration of ships propelled by the screw is greater than in those using the paddle, though the latter roll more in stormy weather.

Twin Screws.—Twin screws are simply two screws, one on each side of the rudder, instead of one screw in the dead wood in front of the rudder. One screw turns to the right and the other to the left. It is claimed for this arrangement, that the ship can be very quickly turned within a small space.

Two-bladed screws are claimed to be more efficient than those with three or four blades, but repeated experi-

ments have shown that, in point of efficiency, there is very little difference between them. Two-bladed screws should be made with a shorter pitch than those having three or four blades.

TABLE

OF THE PROPER PROPORTIONS OF SCREW-PROPELLERS.

Screws of Two Blades.		Screws of Four Blades.		Screws of Six Blades.	
Ratio of Pitch to Diameter.	Fraction of Pitch.	Ratio of Pitch to Diameter.	Fraction of Pitch.	Ratio of Pitch to Diameter.	Fraction of Pitch.
1.006	0.454	1.342	0.454	1.677	0.794
1.069	0.428	1.425	0.428	1.771	0.749
1.135	0.402	1.513	0.402	1.891	0.703
1.205	0.378	1.607	0.378	2.009	0.661
1.279	0.355	1.705	0.355	2.131	0.621
1.357	0.334	1.810	0.334	2.262	0.585
1.450	0.313	1.933	0.313	2.416	0.548
1.560	0.294	2.080	0.294	2.600	0.515
1.682	0.275	2.243	0.275	2.804	0.481

MEASUREMENT OF THE SCREW-PROPELLER.

The surface of a screw-propeller is the same as would be generated by a line revolving around a cylinder, through the axis of which it passes, and at the same time advancing along the axis. In this way the under or back surfaces of the blade may be supposed to be formed, and then the proper thickness is put on, so as to make the front or entering surfaces.

All measurements of a blade should, of course, be made on the back surface. It will be evident, from the explanation of the manner in which the surface of a blade is formed, that by varying the shape of the generating line, or at the rate of its motion along the axis, very different forms of blades can be produced. The pitch of a screw is the distance the generating line moves in the direction of

the axis while it is making one revolution around the cylinder.

It is evident from this that the pitch of the screw may be constant throughout, or it may vary from forward to after part of the blade, or from hub to periphery, according to the rate of motion of the generating line in an axial direction, and its angle of inclination to the axis. Hence, in measuring a screw-propeller, it will be necessary to determine the pitch at a number of points, for the purpose of ascertaining whether it is variable or constant.

Every point in the generating line describes a curve which is called a helix. If measurements are taken along one of these helices, they will show whether the pitch varies from forward to after part of the blade, and measurements on corresponding points of different helices will indicate whether or not the pitch is constant from hub to periphery. As a general thing, the hub of a screw-propeller is faced off at the ends, and the blades do not overhang a plane passing through this face. If necessary, however, a faced surface can be fitted to the hub, and made thick enough for its plane to clear the blades.

Provide a straight-edge a little longer than the radius of the propeller, and secure cleats to it at every foot of its length for large wheels, and from six to nine inches apart for small wheels. These cleats are intended to serve as guides for a rule, so that measurements can be made with accuracy at right angles to the straight-edge. Secure to the end of the hub a piece of paper on which the centre of the hub is marked, and the circumference is divided into any number of equal parts.

Then place the straight-edge on the end of the hub, bringing a mark near its end to the centre of the hub, and making its direction coincide with a division of the circumference. Measure the perpendicular distance from

the straight-edge to the surface of the blade at each one of the cleats; then move the straight-edge to coincide with the next division of the circumference, and again take measurements.

Suppose that the circumference of the hub be divided into thirty-two equal parts, and that the measurements from the straight-edge to the blade, taken at each cleat, are each six inches, then move the straight-edge to the next position, and suppose that the measurements are each fourteen inches. This shows that the generatrix, in one thirty-second of a revolution, has advanced eight inches in an axial direction; consequently, the pitch is thirty-two times as much, or twenty-one feet and four inches.

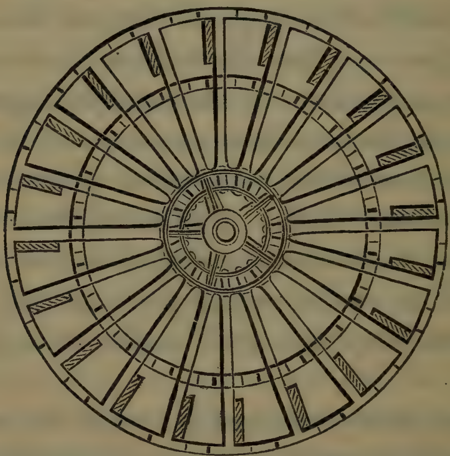
If measurements taken as successive divisions of the circumference give a successive increase of eight inches for each division, it shows that the propeller is a true screw with a pitch of twenty-one feet and four inches.

It will be observed that the measurements made at one cleat in different positions of the straight-edge give determination for the pitch at different points of the same helix, and therefore show whether the pitch varies from forward to after part of the blade. The measurements taken at different cleats, in successive positions of the straight-edge, show the pitch at corresponding points of different helices, and indicate whether the pitch varies from hub to periphery.

The method here described is one of the simplest and most accurate that can be given for determining the pitch of a screw-propeller. The other measurements—the diameter of screw, length of blade, dimensions of hub, and fraction of pitch employed—are so simple as to need no explanation.

HOW TO LINE UP A PROPELLER-SHAFT.

Put two straight-edges on the slides, one at each end; run a line through their centre points, and continue it beyond the shaft. Set a **T** square on one of the straight-edges, making one edge of the blade cut the centre point. Then erect a perpendicular, at the centre of the shaft to the line previously run, by looking it out of wind with the edge of the **T** square, or arranging it so that, when viewed from a distance, it covers the edge of the **T** square for the whole length. Then disconnect the crank from the rod, and swing it on the centre and half-centre, and measure the distances on its face and the two lines. If they vary at different points, the shaft is not in line, and must be adjusted until the distances are the same from all points of the revolution.



Radial Wheel.

PADDLE-WHEELS.

Paddle-wheels consist of two large wheels moving on the end of the engine-shaft. They are made by attaching arms to the centres on the shaft and to two large rings, on

which are bolted the paddles or floats. As they are turned round, the resistance offered to them by the water causes the vessel to move, acting precisely on the same principle as a boat-oar; by them the inertia of the water is made a means of locomotion.

In using this appliance as a motive-power, its advantage greatly depends upon the amount of immersion. When the water approaches the centre, or reaches above, it is obvious the greatest waste of power will ensue. It is quite as obvious that the greater the diameter of the wheel the greater the leverage, and the greater is the effect obtained. There are various kinds of paddle-wheels, such as the ordinary radial, the cycloidal, and the feathering.

The Ordinary Radial Wheel.—This wheel has the floats fixed on the radial arms. In this arrangement the floats enter the water with the whole of their faces presented to it; the same action takes place as they come out. From this arises a great loss of power, for they should evidently offer the greatest resistance to the water when at their lowest point, and none when entering or leaving. From this cause, and the yielding of the water, the ship does not move as fast as the wheel. The loss is called slip, and is generally allowed to be from 10 to 20 per cent.

Cycloidal Wheels.—To obviate the difficulties and disadvantages of the ordinary radial wheel, the cycloidal was advocated. Its peculiarity consists in dividing the float into two strips longitudinally. The strip farthest from the centre is behind the radius, and the other in front of it. The intention of this arrangement is, that the floats may meet the water with more uniformity. It is a very good form of wheel for large vessels. In order that the floats may enter and leave the water with the least possible resistance, they should enter in a tangential direction

to the curve which is being described by any point in the wheel. This is what is known as the cycloidal curve.

Feathering Wheel.

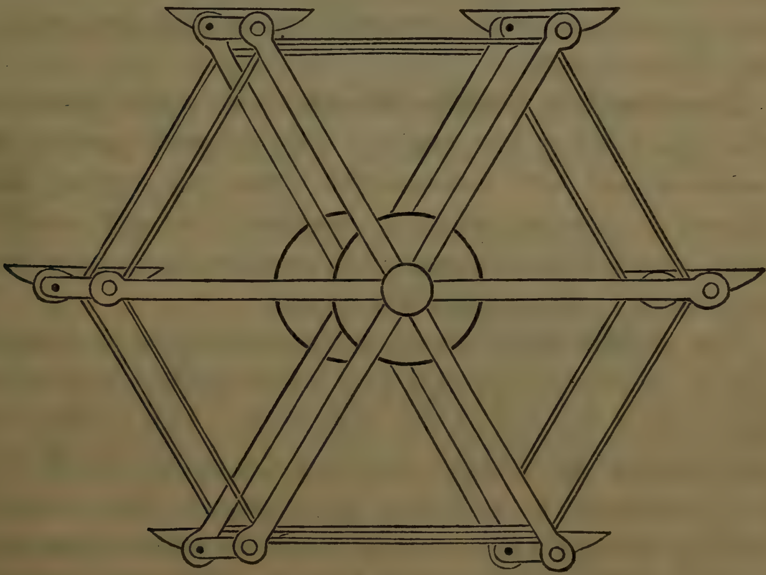
The Feathering Wheel.— In the feathering wheel the floats are governed by mechanism, which causes them to enter and leave the water in a position perpendicular to the direction of their motion. By this arrangement they offer the greatest resistance at the lowest point; the floats are in fact at right angles to the surface of the water when immersed.

In the feathering wheel, as each float is always perpendicular to the water, they progress with the same horizontal velocity, therefore the point of maximum resistance or centre of pressure must be in a line passing longitudinally along the centre of the float. But in the radial wheel this cannot be the case, for the outside edge of the float moves much faster than the inside; the point where these two average each other is taken at a distance of one-third the depth of the board from the outer edge.

Although the feathering wheel produces more useful

effect with the same application of power than the radial wheel, there are many practical objections to its use, the most prominent of which are, the increased first cost, the excessive weight, and the frequent overhauling that they require.

The Manley Paddle-Wheel.—This wheel has only five or six floats, each of which is secured to a rock-shaft, to which a crank is attached. The feathering mechanism is a frame eccentric to the main shaft, connected with each of the cranks by an arm. Each float is secured to the rock-shaft below the centre, so as to divide the pressure on the float equally between the feathering mechanism and the adjacent side frame of the wheel. It is the first machine for marine propulsion which, by its action and application of power, has imitated the Indian's paddle, and has conformed to the first great principle necessary to be observed in propelling a vessel through the water, by obtaining the proper resistance for the power upon the water.



Manley Paddle-Wheel.

This wheel does for the steam-ship what the Indian's paddle does for his boat, and even in greater perfection. It drops a paddle into undisturbed water, forces it backward or forward, as the case may be, in a direction exactly perpendicular to the line of flotation, and, as it is being withdrawn from the water, another paddle is entering far ahead and grasping resistance entirely unused by the preceding paddle.

The operation is certain and constant while the power is applied.

Immersion of Paddles.—The great difficulty with paddle-wheels is to secure a proper immersion. As the ship proceeds on its voyage and consumes its store of coal, the vessel becomes lighter, and, consequently, its draught of water decreases. Therefore, supposing a paddle is properly immersed at the commencement of a voyage, it will be nearly out of the water at the end. At the commencement of a voyage the paddle must be too deeply immersed, at the middle the proper immersion will perhaps be attained, while there will be too little towards the end of the voyage.

Disconnecting Paddle-Wheels.—In some instances, when the wind is fair and the ship is under canvas, the paddle-wheels are disconnected from the engine, and allowed to revolve on their bearings. Several contrivances, which it is unnecessary to mention here, have been introduced to accomplish this object.

The paddle-shaft, where it passes through the vessel's side, is usually surrounded with a lead stuffing-box, which will yield if the end of the shaft falls. This stuffing-box prevents leakage into the ship from the paddle-wheels; but it is expedient, as a further precaution, to have a small tank on the ship's side a little below the stuffing-box, with a pipe leading down to the bilge, to catch and conduct away any water that may enter.

FLUID RESISTANCE.

The scientific investigation of bodies in moving through a fluid is still involved in much obscurity, from the want of independent research on the part of the various authors who have undertaken the elucidation of the subject; and the mistakes incidental to the researches of Newton, and other eminent philosophers, have overrun various departments of physical science, and are now found most difficult of eradication. The circumstances in connection make it expedient to investigate the matter in a practical way, and to illustrate a few of the leading principles of mechanics which relate to this question.

Mechanical power is pressure acting through space; and the amount of mechanical power developed by any combination is measurable by the amount of the pressure multiplied by the amount of space through which the pressure acts. A pressure of 10 pounds acting through a space of 1 foot represents the same amount of mechanical power as a pressure of 1 pound acting through a space of 10 feet; and 10 pounds gravitating through 1 foot, or 1 pound gravitating through 10 feet, represent ten times the amount of mechanical power due to the gravitation of 1 pound through 1 foot.

In the same way, 1000 pounds gravitating through 1 foot is equivalent to 1 pound gravitating through 1000 feet, and, in general terms, the weight or pressure multiplied by the space through which it acts represents the power universally. If, therefore, a body falls freely through space by the operation of gravity, since it parts with none of its power during its descent, the whole power must be accumulated in the falling body in the shape of momentum; and, at the instant of reaching the ground, the body must have such an amount of mechanical power

stored up in it, as would suffice to carry it up again to the position from which it fell, if the power were directed to the accomplishment of that object.

The amount of mechanical power, therefore, in any moving body, is measurable by the weight of the body multiplied by the space through which it must have fallen by gravity, to acquire the velocity it possesses; and this fundamental law, if distinctly apprehended, and kept constantly in recollection, will insure exemption from the fallacies which prevail so generally among authors in reference to such subjects. In Newton's "Second Law of Motion," it is maintained that "the change or alteration of motion, produced in a body by the action of any external force, is always proportional to that force," from whence it is inferred, that to produce twice the quantity of motion in a body, will require just twice the power; and this is the doctrine maintained by Robinson in his "Mechanical Philosophy," and by Hutton, Gregory, and most other English authors who have undertaken to illustrate such questions.

Nevertheless, there is no doubt whatever that this doctrine is altogether erroneous, as was shown by Leibnitz at the time of its promulgation, and subsequently by Smeaton, who, by a series of carefully executed experiments, proved very clearly that, to double the velocity of a moving body, it required four times the amount of mechanical power that was necessary to put it into motion at first; and consequently, that the momentum of moving bodies of the same weight varies as the squares of their respective velocities.

The soundness of this conclusion is made manifest by a reference to the law of falling bodies, by which it will be found that it is necessary a body should fall through four times the height to double its ultimate speed; nine

times the height, to treble its ultimate speed, and so on; showing that the height, and therefore the power exerted in creating the motion, must be as the square of the ultimate speed; and consequently, that the ultimate velocities of all falling bodies will be as the square roots of the heights from which they have respectively descended. In the case of two bodies of equal weight, therefore, moving in space, but of which one moves with twice the velocity of the other, the faster will have four times the amount of mechanical power stored up in it that is possessed by the slower; for it must have fallen from four times the height, to acquire its doubled velocity; and the relative quantities of power capable of being exerted by bodies of the same weight are measurable in all cases by the spaces through which the weight or pressure acts.

A cannon-ball moving with a velocity of 2000 feet a second, has four times the momentum of a cannon-ball of equal weight moving with a velocity of 1000 feet a second; and every particle of a stream of water moving with a velocity of 10 miles an hour has four times the momentum of every particle of a stream of water with a velocity of five miles an hour. Every particle of the faster stream, therefore, will exert four times the effect in impelling any body on which it impinges, that is exerted by every particle of the slower stream. But in the faster stream not only will every particle impinge with four times the force, but there will be twice the number of particles impinging in a given time, and a quadrupled force for each particle; and twice the number of particles striking in a given time gives an effect eight times greater, in a given time, with a doubled velocity of the stream.

Accordingly, it is found that in a water- or wind-mill, when the velocity of the current is doubled, the power exerted is about eight times greater than before; and it is

also found that a steam-vessel, to realize a double velocity, requires about eight times the amount of power. But these results, it is obvious, have reference, not merely to the increased velocity of the particles of matter, but to the large number of them brought into operation; and any given quantity of water, if flowing with a doubled velocity, would only exert four times the power exerted before. In the same manner, a steam-vessel, to accomplish any given voyage in half the time, would require four times the quantity of coal previously consumed; for although eight times the quantity of coal would be consumed per hour, yet only half the number of hours would be occupied in accomplishing the distance.

The number of particles of water to be displaced by a vessel in performing any given voyage is the same, whatever the velocity of the vessel may be; but the number of particles displaced in the hour differs with every different velocity, and the power expended must consequently vary in a corresponding proportion. It may, hence, be asserted generally, that the power or dimension of an engine necessary to propel a vessel increases nearly as the cube of the velocity required to be attained; but the consumption of fuel will only increase in about the ratio of the square of the velocity, looking to the number of miles of distance actually performed by a steamship.

In order to be able to calculate the absolute amount of power required to produce a given effect, it is necessary to be acquainted with the laws which govern the resistance of fluids to the motion of solid bodies in them, which are generally admitted to be based on the following theorem. If a plain surface move at a given velocity through a fluid at rest, in a direction perpendicular to itself, the resistance is proportional to the density of the fluid, and to the square of the velocity of the plane.

SIGNIFICATION OF SIGNS USED IN CALCULATIONS.

$=$	signifies	Equality,	as	3 added to 2 $= 5$.
$+$	"	Addition,	"	$4 + 2 = 6$.
$-$	"	Subtraction,	"	$7 - 4 = 3$.
$\times$	"	Multiplication,	"	$6 \times 2 = 12$.
$\div$	"	Division,	"	$16 \div 4 = 4$.
: :: :	"	Proportion,	"	2 is to 3, so is 4 to 6.
$\sqrt{}$	"	Square Root,	"	$\sqrt{} = 4$.
$\sqrt[3]{}$	"	Cube Root,	"	$\sqrt[3]{64} = 4$.
3^2	"	3 is to be squared,	"	$3^2 = 9$.
3^3	"	3 is to be cubed,	"	$3^3 = 27$.

$2 + 5 \times 4 = 28$ signifies that two, three, or more numbers are to be taken together, as $2 + 5 = 7$, and 4 times $7 = 28$.

$\sqrt{5^2 - 3^2} = 4$ signifies that 3 squared taken from 5 squared, and the square root extracted $= 4$.

$\sqrt[3]{\frac{10 \times 6}{15}} = 1{\cdot}587$ signifies that where 10 is multiplied by 6 and divided by 15, the cube root of the quotient $= 1{\cdot}587$.

DECIMAL.

Decimal Arithmetic is of Hindoo origin, and was introduced into Arabia about one thousand years ago, from whence it spread throughout Europe and the entire civilized world. The base, 10, originated from the ten fingers, which were used for counting before characters were formed to denote numbers. The base, 10, admits of only one binary division, which gives the prime number 5 without fraction. The trinary divisions give an endless number of decimals.

Decimal Fractions are fractions in which the denominator is a unit or 1 with ciphers annexed, in which case

they are commonly expressed by writing the numerator only with a point before it, by which it is separated from whole numbers; thus, ·5, which denotes five-tenths, $\frac{5}{10}$; ·25, that is, $\frac{25}{100}$.

DECIMAL EQUIVALENTS OF INCHES, FEET, AND YARDS.

Fractions of an Inch.		Decimals of an Inch.		Decimals of a Foot.
	—	·0625	=	·00521
$\frac{1}{8}$	—	·125	=	·01041
	—	·1875	=	·01562
$\frac{1}{4}$	—	·25	=	·02083
	—	·3125	=	·02604
$\frac{3}{8}$	—	·375	=	·03125
	—	·4375	=	·03645
$\frac{1}{2}$	—	·5	=	·04166
	—	·5625	=	·04688
$\frac{5}{8}$	—	·625	=	·05208
	—	·6875	=	·05729
$\frac{3}{4}$	—	·75	=	·06250
	—	·8125	=	·06771
$\frac{7}{8}$	—	·875	=	·07291
	—	·9375	=	·07812
1 inch	—	1·00	=	·08333

Inch.		Feet.		Yards.
1	=	·0833	=	·0277
2	=	·1666	=	·0555
3	=	·25	=	·0833
4	=	·3333	=	·1111
5	=	·4166	=	·1389
6	=	·5	=	·1666
7	=	·5833	=	·1944
8	=	·6666	=	·2222
9	=	·75	=	·25
10	=	·8333	=	·2778
11	=	·9166	=	·3055
12	=	1·000	=	·3333

DECIMAL EQUIVALENTS OF POUNDS AND OUNCES.

Oz.	Lbs.	Oz.	Lbs.	Oz.	Lbs.	Oz.	Lbs.	Oz.	Lbs.
$\frac{1}{4}$	·015625	3	·1875	$6\frac{1}{2}$	·40625	10	·625	$13\frac{1}{2}$	·84375
$\frac{1}{2}$	·03125	$3\frac{1}{2}$	·21875	7	·4375	$10\frac{1}{2}$	·65625	14	·875
$\frac{3}{4}$	·046875	4	·25	$7\frac{1}{2}$	·46875	11	·6875	$14\frac{1}{2}$	·90625
1	·0625	$4\frac{1}{2}$	·28125	8	·5	$11\frac{1}{2}$	·71875	15	·9375
$1\frac{1}{2}$	·09375	5	·3125	$8\frac{1}{2}$	·53125	12	·75	$15\frac{1}{2}$	·96875
2	·125	$5\frac{1}{2}$	·34375	9	·5625	$12\frac{1}{2}$	·78125	16	1·
$2\frac{1}{2}$	·15625	6	·373	$9\frac{1}{2}$	·59375	13	·8125		

USEFUL NUMBERS IN CALCULATING WEIGHTS AND MEASURES, ETC.

Feet	multiplied by	.00019	equals	miles.
Yards	"	·0006	"	miles.
Links	"	·22	"	yards.

Links	multiplied by	·66	equals	feet.
Feet	"	1·5	"	links.
Square inches	"	·007	"	square feet.
Circular inches	"	·00546	"	square feet.
Square feet	"	·111	"	square yards.
Acres	"	·4840	"	square yards.
Square yards	"	·0002066	"	acres.
Width in chains	"	·8	"	acres per m.
Cube feet	"	·04	"	cube yards.
Cube inches	"	·00058	"	cube feet.
U. S. bushels	"	·0495	"	cube yards.
U. S. bushels	"	1·2446	"	cube feet.
U. S. bushels	"	2150·42	"	cube inches.
Cube feet	"	·8036	"	U. S. bushels.
Cube inches	"	·000466	"	U. S. bushels.
U. S. gallons	"	·13367	"	cube feet.
U. S. gallons	"	·231	"	cube inches.
Cube feet	"	7·48	"	U. S. gallons.
Cylindrical feet	"	5·874	"	U. S. gallons.
Cube inches	"	·004329	"	U. S. gallons.
Cylindrical inches	"	·0034	"	U. S. gallons.
Pounds	"	·009	"	cwt.
Pounds	"	·00045	"	tons.
Cubic foot of water	"	62·5	"	lbs. avoird.
Cubic inch of water	"	·03617	"	lbs. avoird.
Cylindrical foot of water	"	49·1	"	lbs. avoird.
Cylindrical inch of water	"	·02842	"	lbs. avoird.
U. S. gallons of water	"	13·44	"	1 cwt.
U. S. gallons of water	"	268·8	"	1 ton.
Cubic feet of water	"	1·8	"	1 cwt.
Cubic feet of water	"	35·88	"	1 ton.
Cylindrical foot of water	"	6·	"	U. S. gallons.
Column of water, 12 in. high, 1 in. diameter			"	341 lbs.
183·346 circular inches			"	1 square foot.
2200 cylindrical inches			"	1 cubic foot.
French metres	multiplied by	3·281	"	feet.
Kilogrammes	"	2·205	"	avoird. lbs.
Grammes	"	·002205	"	avoird. lbs.

DECIMAL EQUIVALENTS TO THE FRACTIONAL PARTS OF A GALLON OR AN INCH.

(The Inch or Gallon being divided into 32 parts.)

(In multiplying decimals, it is usual to drop all but the first two or three figures.)

Decimals.	Gallon or Inch.	Gills.	Pints.	Quarts.	Decimals.	Gallon or Inch.	Gills.	Pints.	Quarts.	Decimals.	Gallon or Inch.	Gills.	Pints	Quarts.
·03125	1–32	1	¼	⅛	·375	3–8	12	3	1½	·71875	23–32	23	5¾	2⅞
·0625	1–16	2	½	¼	·40625	13–32	13	3¼	1⅝	·75	3–4	24	6	3
·09375	3–32	3	¾	⅜	·4375	7–16	14	3½	1¾	·78125	25–32	25	6¼	3⅛
·125	1–8	4	1	½	·46875	15–32	15	3¾	1⅞	·8125	13–16	26	6½	3¼
·15625	5–32	5	1¼	⅝	·5	½	16	4	2	·84375	27–32	27	6¾	3⅜
·1875	3–16	6	1½	¾	·53125	17–32	17	4¼	2⅛	·875	7–8	28	7	3½
·21875	7--32	7	1¾	⅞	·5625	9–16	18	4½	2¼	·90625	29–32	29	7¼	3⅝
·25	1–4	8	2	1	·59375	19–32	19	4¾	2⅜	·9375	15–16	30	7½	3¾
·28125	9–32	9	2¼	1⅛	·625	5–8	20	5	2½	·96875	31–32	31	7¾	3⅞
·3125	5–16	10	2½	1¼	·65625	21–32	21	5¼	2⅝	1·000	1	32	8	4
·34375	11–32	11	2¾	1⅜	·6875	11–16	22	5½	2¾					

UNITS.

Unit of Heat. — The unit of heat varies: the French unit of heat, called a "*caloric*," is the amount of heat necessary to raise one kilogramme (2·2046215 pounds) of water one degree Centigrade, or from 0° C. to 1° C. In this country and in England the amount of heat necessary to raise one pound of water one degree Fahrenheit, or from 32° Fah. to 33° Fah., is taken as the unit of heat.

Unit of Length. — The unit of length used in this country and in England is the yard, the length of which has been determined by means of a pendulum, vibrating seconds in the latitude of London, in a vacuum and at the level of the sea. The length of such a pendulum is to be divided into 3,913,929 parts, and 3,600,000 of these parts are to constitute a yard. The yard is divided into 36 inches, so that the length of the seconds pendulum in London is 39·13929 inches.

The French unit of length, called the mètre, has been taken as being the ten-millionth part of the quadrant of a meridian passing through Paris; that is to say, the ten-millionth part of the distance between the equator and the pole, measured through Paris. It is equal to 39·3707898 inches. The mètre is divided into one thousand millimètres, one hundred centimètres, and ten decimètres; while a decamètre is ten mètres; a hectomètre one hundred mètres, a kilomètre one thousand mètres, and a myriamètre ten thousand mètres. The following table gives the value of these measurements in English inches and yards:

	In English Inches.	In English Yards.
Millimètre	0·03937	0·0010936
Centimètre	0·39371	0·0109363
Decimètre	3·93708	0·1093633
Mètre	39·37079	1·0936331
Decamètre	393·70790	10·9363310
Hectomètre	3937·07900	109·3633100
Kilomètre	39370·79000	1093·6331000
Myriamètre	393707·90000	10936·3310000

One English yard is equal to 0·91438 mètre; while one mile is equal to 1·60931 kilomètres.

Unit of Surface. — For the unit of surface, the square inch, foot, and yard adopted in this country and in England are replaced in the metric system by the square millimètre, centimètre, decimètre, and mètre.

1 square mètre = 1·1960333 square yards.
1 square inch = 6·4513669 square centimètres.
1 square foot = 9·2899683 square decimètres.
1 square yard = 0·83609715 square mètre.

Unit of Capacity. — The cubic inch, foot, and yard furnish measures of capacity; but irregular measures, such as the pint and gallon, are also used in this country

and in England. The gallon contains ten pounds avoirdupois weight of distilled water at 62° Fah.; the pint is one-eighth part of a gallon.

The French unit of capacity is the cubic decimètre or litre, equal to 1·7607 English pints, or 0·2200 English gallon; and we have cubic inches, decimètres, centimètres, and millimètres.

1 litre	=	61·027052 cubic inches.
1 cubic foot	=	28·315311 litres.
1 cubic inch	=	16·386175 cubic centimètres.
1 gallon	=	4·543457 litres.

Unit of Weight.—The unit of weight used in this country and in England—the pound—is derived from the standard gallon, which contains 277·274 cubic inches; the weight of one-tenth of this is the pound avoirdupois, which is divided into 7000 grains.

The French measures of weight are derived at once from the measures of capacity, by taking the weight of cubic millimètres, centimètres, decimètres, or mètres of water at its maximum density, that is at 4° C. or 39° Fah. A cubic mètre of water is a tonne, a cubic decimètre a kilogramme, a cubic centimètre a gramme, and a cubic millimètre a milligramme.

	In English Grains.	In Pounds Avoirdupois 1 pound = 700 grammes.
Milligramme ($\frac{1}{1000}$ part of a gramme).	0·015432	0·0000022
Centigramme ($\frac{1}{100}$ part of a gramme)..	0·154323	0·0000220
Décigramme ($\frac{1}{10}$ part of a gramme)....	1·543235	0·0002205
Gramme..	15·432349	0·0022046
Décagramme (10 grammes)................	154·323488	0·0220462
Hectogramme (100 grammes)............	1543·324880	0·2204621
Kilogramme (1000 grammes)............	15432·348800	2·2046213
Myriagramme (10000 grammes)........	154323·488000	22·0462126

Unit of Time or Duration.—The unit of time or dura-

tion is the same for all civilized countries. The twenty-fourth part of a mean solar day is called an hour, and this contains sixty minutes, each of which is divided into sixty seconds. The second is universally used as the unit of duration.

Another unit of time is the period occupied by the earth in making one revolution around the sun, in reference to an assumed fixed star, which unit is called a sidereal year, and contains 365 days, 6 hours, 9 minutes, and 9·6 seconds mean solar time.

Unit of Velocity.—The units of velocity adopted by different scientific writers vary somewhat; the most usual, perhaps, in regard to sound, falling bodies, projectiles, etc., is the velocity of feet or mètres per second. In the case of light and electricity, miles and kilomètres per second are employed.

Unit of Work.—In this country and in England the unit of work is usually the foot-pound, viz., the force necessary to raise one pound weight one foot above the earth in opposition to the force of gravity. A horse-power is equal to 33,000 pounds raised to a height of one foot in one minute of time.

In France the kilogrammètre is the unit of work, and is the force necessary to raise one kilogramme to a height of one mètre against the force of gravity. One kilogrammètre = 7·233 foot-pounds. The cheval-vapeur is nearly equal to the English horse-power, and is equivalent to 32,500 pounds raised to a height of one foot in one minute of time. The force competent to produce a velocity of one mètre in one second, in a mass of one gramme, is sometimes adopted as a unit of force.

Unit of Pressure.—The pressure of the atmosphere at the level of the ocean, with the barometer at 30 inches, is taken as the unit in estimating and comparing pressures and elastic forces.

THEORY OF THE STEAM-ENGINE.

WATER.

AIR. THERMOMETERS. ELASTIC FLUIDS.

CALORIC.

HEAT. COMBUSTION. GASES.

STEAM.

WATER.

The Composition of Water.—Pure water is composed of the two gases, hydrogen and oxygen, in the proportions of 2 measures of hydrogen to 1 of oxygen, or, 1 weight of hydrogen to 8 of oxygen; or, oxygen 89 parts by weight, and by measure 1 part, hydrogen, by weight, 11 parts, and by measure 2 parts.

Pure water in nature does not exist, nor is it to be found in the laboratory of the chemist. Fortunately, however, it happens that pure water is not necessary, or even desirable, for household or manufacturing purposes. The presence of air or other gases adds greatly to the ease with which steam may be generated; the ammonia that is present in most water improves it for manufacturing purposes, and it has been abundantly proved that the salts which are present in most well-waters add greatly to their wholesomeness.

But at the same time it must be remembered that some waters contain impurities which render them unfit for use. Of these various impurities, the insoluble portion is in general the least injurious, though it is frequently the most offensive.

Water swarming with minute animalcules, or turbid with the clay and sand that have been stirred up from the bed of some stream, may be offensive though it is not dangerous; while, on the other hand, water may be beautifully clear to the eye and not very offensive to the taste, and yet hold in solution the most deadly poison, in the form of dissolved salts or the soluble portions of animal excreta.

It also happens that these insoluble matters are easily and cheaply removed, while the utmost care is required to free water from matter which exists in a dissolved state.

NAYLOR'S VERTICAL ENGINE.

The specific gravity of all waters is not the same. The following table will show the specific gravity of different seas.

	Weight of water being 1000.	Weight of an imperial gallon in pounds.
Water from the Dead Sea	1240	12.4
" " " Mediterranean	1029	10.3
" " " Irish Channel	1028	10.2
" " " Baltic Sea	1015	10.2

For the production of steam, all waters are not equal. Water holding salt in solution, earth, sand or mud in suspension, requires a higher temperature to produce steam of the same elastic force than that generated from pure water.

Water, like all other fluids and gases, expands with heat and contracts with cold down to 40° Fah.

If water be boiled in an open vessel it is impossible to raise the temperature above 212° Fah., as all the surplus heat which may be applied passes off with the steam.

If heat be applied to the top of a vessel, ebullition will not take place, as very little heat would be communicated to other parts of the vessel, and the water would not boil.

Ebullition, or boiling of water or other liquids, is effected by the communication of heat through the separation of their particles.

The evaporation of water is the conversion of water as a liquid into steam as a vapor.

Latent Heat of Water or Ice.—If a pound of ice at 32° Fah. be mixed with a pound of water at 111°, the water will gradually dissolve the ice, being just sufficient for that purpose, and the residuum will be two pounds of water 32° Fah.

The 79 units of heat which are apparently lost having been employed in performing a certain amount of work, *i. e.*,* in melting the ice or separating the molecules and giving them another shape, and as all work requires a supply of heat to do it, these 79 units have been consumed in performing the work necessary to melt the ice.

Latent Heat of Water.—If the pound of water were reconverted into ice, it would have to give up the 79 units of latent heat. Hence we see why it should be called the latent heat of water and not the latent heat of ice.

Suppose that we have a pound of ice, at a temperature of 32° Fah., and that we mix it with a pound of water at 212°, the ice will be melted and we shall have two pounds of water at a temperature of 51°.

Now, if we take a pound of ice at a temperature of 32° and mix it with a pound of water at 212°, the resulting mixture of the two pounds will have a temperature of 122°. Hence we see that the ice, in melting, has absorbed enough heat to raise two pounds of water through a temperature of 122°—51°=71°, or one pound through 142°, and we say that the latent heat of the liquefaction of water is 142°.

The latent heat of the evaporation of water can be determined in a similar manner by condensing a pound of steam at 212° Fah. with a given weight of water at a known temperature, and also by mixing a pound of water at a temperature of 212° Fah. with the same amount of water as was employed in the case of steam, and observing the difference of temperature of the resulting mixtures.

Thus, a pound of water at 212° mixed with ten pounds at 60° gives eleven pounds at 74°. A pound of steam at 212° mixed with ten pounds of water at 60° gives eleven pounds of water at 162°. In other words, the steam on

* *i. e.*, that is.

being condensed has given out heat (which was not previously sensible to the thermometer) enough to raise eleven pounds of water through a temperature of 162° less 74° equals 88°, or one pound through 968°, and we say that the latent heat of the evaporation of water is 968°.

If a pound of mercury and a pound of water be heated to the same temperature, and allowed to cool, it will be found that the mercury cools 30 times as fast as the water; hence we say that the specific heat of mercury is about one-thirtieth that of water.

The boiling-point of water is that temperature at which the tension of its vapor exactly balances the pressure of the atmosphere. But the temperature at which the ebullition of water begins depends upon the elasticity of the air or other pressure.

At the level of the sea, the barometer standing at 29·905 (or nearly 30) inches of mercury, water will boil at 212° Fah.; but the higher we ascend above the level of the sea, the more the boiling-point diminishes.

Water attains its greatest density at 39° Fah., or 7° above freezing.

Although it is claimed that water presses in every direction and finds its level, water can be compressed $\frac{1}{100}$ of an inch in every 18 feet by each atmosphere or pressure of 15 pounds to the square inch of pressure applied; but when the pressure is removed, its elasticity restores it to its original bulk.

Water becomes solid and crystallized as ice owing to the abstracting of its heat.

The force of expansion exerted by water in the act of freezing has been found irresistible in all mechanical experiments to prevent it.

Water in a vacuum boils at about 98° Fah., and assumes a solid at 32 degrees in the atmosphere, when it expands $\frac{1}{17}$ its original bulk.

Water, after being long kept boiling, affords an ice more solid, and with fewer air-bubbles, than that which is formed from unboiled water.

Pure water, kept for a long time in vacuo, and afterwards frozen there, freezes much sooner than common water exposed to the same degree of cold in the open atmosphere.

Ice formed of water thus divested of its air, is much more hard, solid, heavy, and transparent than common ice.

Ice, after it is formed, continues to expand by decrease of temperature; to which fact is probably attributable the occasional splitting and breaking up of the ice on ponds, etc.

A cubic foot of water weighs $62\frac{1}{2}$ pounds; a cubic foot of ice weighs 57·2 pounds. It follows that ice is nearly one-twelfth lighter than water.

Now, if heat be applied to ice, the temperature of which is below freezing, the temperature will soon rise to 32° or freezing; but any further application of heat cannot increase the temperature of the ice until the whole mass is melted.

The specific gravity of ice is ·92, and specific gravity of water is 1000 — water being the standard by which to obtain the specific gravity of all solids, fluids, and even gases. Though air is sometimes used as a standard for gases, water is more commonly used.

The specific gravity of water is the comparative weight of a given bulk of water to the same bulk of any other liquid. Thus, if we take equal measures of the several different liquids, we shall find that they possess very different weights.

The weight of a pint of water, a pint of oil, and a pint of mercury will differ very materially. The mercury will weigh 13 times more than water does, and the water will weigh a good deal more than the oil.

TABLE

SHOWING THE WEIGHT OF WATER.

1	Cubic inch	is equal to	·03617	pounds.
12	Cubic inches	"	·434	"
1	Cubic foot	"	62·5	"
1	Cubic foot	"	7·50	U. S. gallons.
1·8	Cubic foot	"	112·00	pounds.
35·84	Cubic feet	"	2240·00	"
1	Cylindrical inch	"	·02842	"
12	Cylindrical inches	"	·341	"
1	Cylindrical foot	"	49·10	"
1	Cylindrical foot	"	6·00	U. S. gallons.
2·282	Cylindrical feet	"	112·00	pounds.
45·64	Cylindrical feet	"	2240·00	"
11·2	Imperial gallons	"	112·00	"
224·0	Imperial gallons	"	2240 00	"
13·44	U. S. gallons	"	112·00	"
268·8	U. S. gallons	"	2240·00	"

TABLE

SHOWING THE WEIGHT OF WATER AT DIFFERENT TEMPERATURES.

Temperature Fahrenheit.	Weight of a Cubic Foot in Pounds.	Temperature Fahrenheit.	Weight of a Cubic Foot in Pounds.
40°	62.408	172°	60.72
42°	62.406	182°	60.5
52°	62.377	192°	60.28
62°	62.321	202°	60.05
72°	62.25	212°	59.82
82°	62.15	230°	59.37
92°	62.04	250°	58.85
102°	61.92	275°	58.17
112°	61.78	300°	57.42
122°	61.63	350°	55.94
132°	61.47	400°	54.34
142°	61.30	450°	52.70
152°	61.11	500°	51.02
162°	60.92	600°	47.64

Water attains a minimum volume and a maximum density at 40° Fah.; any departure from that temperature in either direction is accompanied by expansion, so that 8° or 10° of cold produce about the same amount of expansion as 8° or 10° of heat.

At 70° Fah., pure water will boil at 1° less of temperature, for an average of about 550 feet of elevation above sea-level, up to a height of one-half of a mile. At the height of 1 mile, 1° of boiling temperature will correspond to about 560 feet of elevation.

The following table shows the approximate altitude above sea-level corresponding to different heights, or readings of the barometer; and to the different degrees of Fahrenheit's thermometer, at which water boils in the open air. Thus, when the barometer, under undisturbed conditions of the atmosphere, stands at 24·08 inches, or when pure rain or distilled water boils at the temperature of 201° Fah., the place is about 5764 feet above the level of the sea, as shown by the table.

TABLE

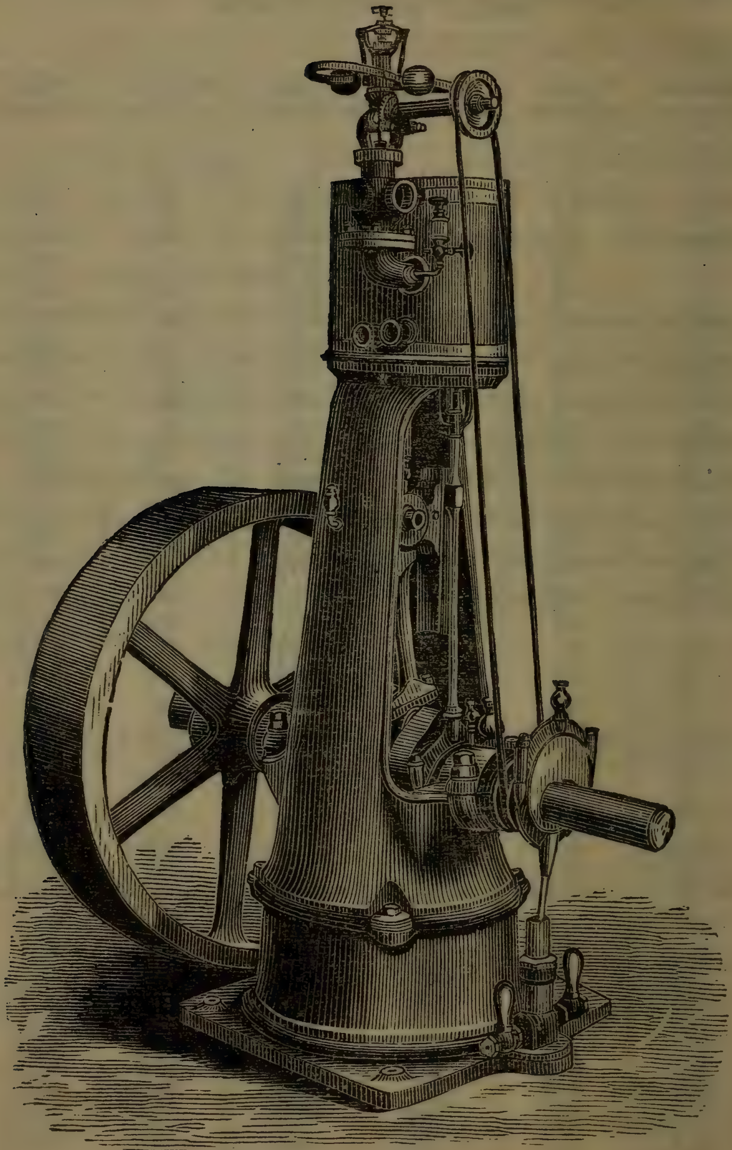
SHOWING THE BOILING-POINT FOR FRESH WATER AT DIFFERENT ALTITUDES ABOVE SEA-LEVEL.

Boiling-point in deg. Fah.	Altitude above sea-level in feet.	Boiling-point in deg. Fah.	Altitude above sea-level in feet.	Boiling-point in deg. Fah.	Altitude above sea-level in feet.
184°	15221	195°	9031	206°	3115
185°	14649	196°	8481	207°	2589
186°	14075	197°	7932	208°	2063
187°	13498	198°	7381	209°	1539
188°	12934	199°	6843	210°	1025
189°	12367	200°	6304	211°	512
190°	11799	201°	5764	212°	sea-level = 0
191°	11243	202°	5225		
192°	10685	203°	4697		
193°	10127	204°	4169	Below sea-level.	
194°	9579	205°	3642	213°	511

TABLE

SHOWING THE WEIGHT OF WATER IN PIPE OF VARIOUS DIAMETERS 1 FOOT IN LENGTH.

Diameter in Inches.	Weight in Pounds.	Diameter in Inches.	Weight in Pounds.	Diameter in Inches.	Weight in Pounds.
3	3	12¼	51	22½	172½
3¼	3½	12½	53¼	23	180¼
3½	4¼	12¾	55½	23½	188¼
3¾	4¾	13	57½	24	196¼
4	5½	13¼	59¾	24½	204½
4¼	6¼	13½	62¼	25	213
4½	7	13¾	64½	25½	222½
4¾	7¾	14	66¾	26	230½
5	8½	14¼	69¼	26½	239½
5¼	9¼	14½	71½	27	248½
5½	10½	14¾	74¼	27½	257¾
5¾	11¼	15	76¾	28	267¼
6	12¼	15¼	79¼	28½	276¾
6¼	13¼	15½	82	29	286½
6½	14½	15¾	84½	29½	296½
6¾	15½	16	87¼	30	306¾
7	16¾	16¼	90	30½	317¼
7¼	18	16½	92¼	31	327½
7½	19¼	16¾	95½	31½	338¼
7¾	20½	17	98½	32	349
8	21¾	17¼	101½	32½	360
8¼	23¼	17½	104½	33	371¼
8½	24½	17¾	107½	33½	382½
8¾	26	18	110½	34	394
9	27½	18¼	113½	34½	405¾
9¼	29¼	18½	116½	35	417½
9½	30¾	18¾	119¾	35½	429½
9¾	32½	19	123	36	441¾
10	34	19¼	126¼	36½	454
10¼	35½	19½	129½	37	466½
10½	37½	19¾	132	37½	479¼
10¾	39¼	20	136¼	38	492¼
11	41¼	20½	143¼	38½	505¼
11¼	44¼	21	150¼	39	518½
11½	45	21½	157½	39½	531¾
11¾	47	22	165	40	545½
12	49				



HASKIN'S VERTICAL HIGH-PRESSURE ENGINE.

AIR.

The atmosphere is known to extend at least 45 miles above the earth.

Its composition is about 79 measures of nitrogen gas and 21 of oxygen; or, in other words, air consists of, by volume, oxygen 21 parts, nitrogen 79 parts; by weight, oxygen 77 parts, nitrogen 23 parts.

According to Dr. Prout, 100 cubic inches of air at the surface of the earth, when the barometer stands at 34 inches, and at a temperature of 60° Fah., weighs about 31 grains, being thus about 815 times lighter than water, and 11,065 times lighter than mercury.

Since the air of the atmosphere is possessed of weight, it must be evident that a cubic foot of air at the surface of the earth has to support the weight of all the air directly above it, and that, therefore, the higher we ascend up in the atmosphere the lighter will be the cubic foot of air; or, in other words, the farther from the surface of the earth, the less will be the density of the air.

At the height of three and a half miles it is known that the atmospheric air is only half as dense as it is at the surface of the earth.

From the nature of fluids, it follows that the atmosphere presses against any body which comes in contact with it, because fluids exert a pressure in all directions — upwards, downwards, sidewise, and oblique.

It is also known that the pressure on any point is equal to the weight of all the particles of the fluid in a perpendicular line between the point in contact and the surface of the fluid.

The amount of pressure of a column of air whose base is one square foot, and altitude the height of the atmosphere, has been found to be 2156 pounds avoirdupois, or

very nearly 15 pounds of pressure on every square inch; consequently, it is common to state the pressure of the atmosphere as equal to 15 pounds on the square inch.

If any gaseous body or vapor, such as steam, exerts a pressure equivalent to 15 pounds on the square inch, then the force of that vapor is said to be equal to one atmosphere; if the vapor be equal to 30 pounds on every square inch, then it is equal to two atmospheres, and so on. Consequently, the atmospheric pressure is capable of supporting about 30 inches of mercury, or a column of water 34 feet high.

It is also found that the pressure of the atmosphere is not constant even at the same place; at the equator, the pressure is nearly constant, but is subject to greater change in the high latitudes.

In some countries the pressure of the atmosphere varies so much as to support a column of mercury so low as 28 inches, and at other times so high as 31, the mean being 29·5, thus making the average pressure between 14 and 15 pounds on the square inch. But in scientific books, generally, the pressure is understood in round numbers to be 15 pounds, so that a pressure exerted equal to 1, 2, 3, 4, etc., atmospheres, means such a pressure as would support 30, 60, 90, 120, etc., inches of mercury in a perpendicular column, or 15, 30, 45, 60, etc., pounds on every square inch.

Air is a very slow conductor of heat, and is sometimes used as a non-conductor in hollow walls to prevent the radiation of heat.

The pressure of the air differs at different latitudes; for instance, at 7 miles above the surface of the earth the air is 4 times lighter than it is at the earth's surface; at 14 miles it is 16 times lighter, and at 21 miles it is 64 times lighter.

Under a pressure of 5½ tons to the square inch, air becomes as dense, and would weigh as much per cubic foot, as water.

The greatest heat of air in the sun is about 140° Fah., and it probably never exceeds 145° Fah.

Air, like all other gases, expands but one volume for each 493° of temperature through which it is raised, and in order to double its volume, we must raise it 493° more, which will bring it to a temperature of 986° Fah.

It requires 13,817 cubic feet of air to make one pound. One cubic foot of air at the surface of the earth weighs 527 grains, or ¼ ounce avoirdupois.

Although the atmosphere may extend to the height of 45 miles, yet its lower half is so compressed as to occupy only 3½ miles, so greatly do the upper portions expand when relieved from pressure. Hence, at the height of 3½ miles, the elasticity of the atmosphere is ½; at 7 miles, ¼; at 10½ miles, ⅛; at 14 miles $\frac{1}{16}$, etc.

TABLE

SHOWING THE WEIGHT OF THE ATMOSPHERE IN POUNDS, AVOIRDUPOIS, ON ONE SQUARE INCH, CORRESPONDING WITH DIFFERENT HEIGHTS OF THE BAROMETER, FROM 28 INCHES TO 31 INCHES, VARYING BY TENTHS OF AN INCH.

Barometer in Inches.	Atmosphere in Pounds.	Barometer in Inches.	Atmosphere in Pounds.	Barometer in Inches.	Atmosphere in Pounds.
28.0	13.72	29.1	14.26	30.1	14.75
28.1	13.77	29.2	14.31	30.2	14.80
28.2	13.82	29.3	14.36	30.3	14.85
28.3	13.87	29.4	14.41	30.4	14.90
28.4	13.92	29.5	14.46	30.5	14.95
28.5	13.97	29.6	14.51	30.6	15.00
28.6	14.02	29.7	14.56	30.7	15.05
28.7	14.07	29.8	14.61	30.8	15.10
28.8	14.12	29.9	14.66	30.9	15.15
28.9	14.17	30.0	14.70	31.0	15.19
29.0	14.21				

TABLE

SHOWING THE EXPANSION OF AIR BY HEAT, AND THE INCREASE IN BULK IN PROPORTION TO INCREASE OF TEMPERATURE.

Fahrenheit.			Bulk.	Fahrenheit.			Bulk.
Temp.	32	Freezing-point.	1000	Temp.	75	Temperate......	1099
"	33	"	1002	"	76	Summer heat..	1101
"	34	"	1004	"	77	"	1104
"	35	"	1007	"	78	"	1106
"	36	"	1009	"	79	"	1108
"	37	"	1012	"	80	"	1110
"	38	"	1015	"	81	"	1112
"	39	"	1018	"	82	"	1114
"	40	"	1021	"	83	"	1116
"	41	"	1023	"	84	"	1118
"	42	"	1025	"	85	"	1121
"	43	"	1027	"	86	"	1123
"	44	"	1030	"	87	"	1125
"	45	"	1032	"	88	"	1128
"	46	"	1034	"	89	"	1130
"	47	"	1036	"	90	"	1132
"	48	"	1038	"	91	"	1134
"	49	"	1040	"	92	"	1136
"	50	"	1043	"	93	"	1138
"	51	"	1045	"	94	"	1140
"	52	"	1047	"	95	"	1142
"	53	"	1050	"	96	Blood heat.....	1144
"	54	"	1052	"	97	"	1146
"	55	"	1055	"	98	"	1148
"	56	Temperate...	1057	"	99	"	1150
"	57	"	1059	"	100	"	1152
"	58	"	1062	"	110	Fever heat 112	1173
"	59	"	1064	"	120	"	1194
"	60	"	1066	"	130	"	1215
"	61	"	1069	"	140	"	1235
"	62	"	1071	"	150	"	1255
"	63	"	1073	"	160	"	1275
"	64	"	1075	"	170	Spirits boil 176	1295
"	65	"	1077	"	180	"	1315
"	66	"	1080	"	190	"	1334
"	67	"	1082	"	200	"	1364
"	68	"	1084	"	210	"	1372
"	69	"	1087	"	212	Water boils	1375
"	70	"	1089	"	302	"	1558
"	71	"	1091	"	392	"	1739
"	72	"	1093	"	482	"	1919
"	73	"	1095	"	572	"	2098
"	74	"	1097	"	680	"	2312

THE THERMOMETER.

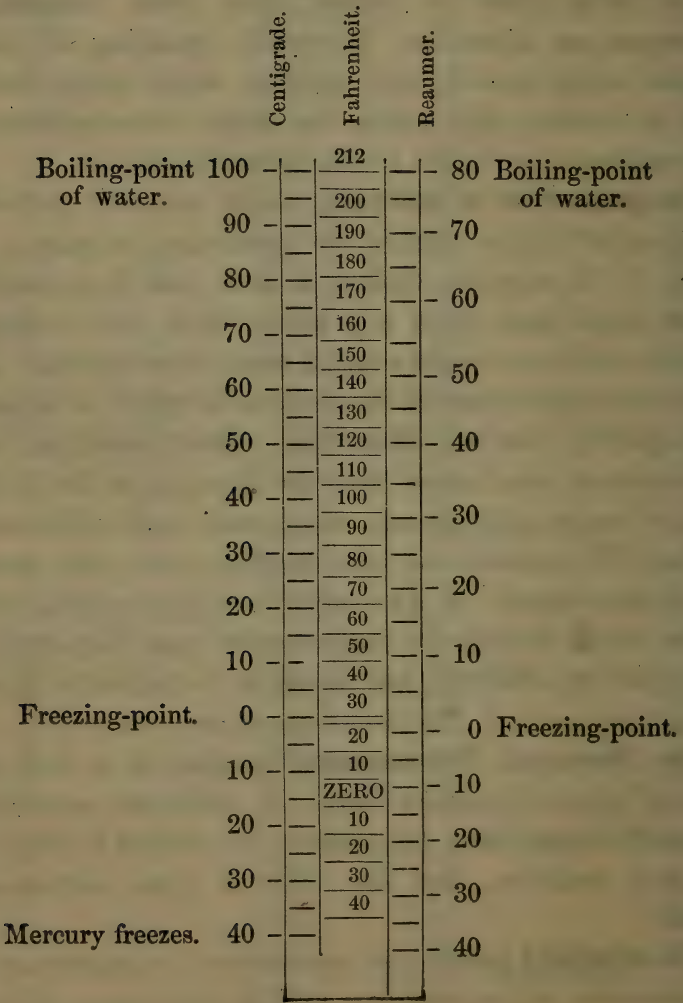
The Thermometer is an instrument for measuring variations of heat or temperature. The principle upon which thermometers are constructed, is the change of volume which takes place in bodies, when their temperature undergoes an alteration. Generally speaking, all bodies expand when heated, and contract when cooled, and in such a manner that under the same circumstances of temperature they return to the same dimensions.

The properties of mercury, which render it preferable to all other liquids (unless for particular purposes), are these: 1. It supports, before it boils and is reduced to vapor, more heat than any other fluid, and endures a greater cold than would congeal most other liquids. 2. It takes the temperature of the medium in which it is placed more quickly than any other fluid. Count Rumford found that mercury was heated from the freezing- to the boiling-point of water in 58 seconds, while water took 133 seconds, and air 617 seconds, the heat applied being the same in all the three cases. 3. The variations of its volume, within limits, which include the temperatures most frequently required to be observed, are found to be perfectly regular and proportional to the variations of temperature.

The Mercurial Thermometer consists of a bulb and stem of glass of uniform bore. A sufficient quantity of mercury having been introduced, it is boiled to expel the air and moisture, and the tube is then hermetically sealed.

The standard points are ascertained by immersing the thermometer in melting ice, and in the steam of water boiling under the pressure of 14·7 pounds on the square inch, and marking the positions of the top of the column; the interval between those points is divided into the proper

COMPARATIVE SCALE OF CENTIGRADE, FAHRENHEIT, AND REAUMER THERMOMETERS.



number of degrees — 100 for the Centigrade scale; 180 for Fahrenheit's; and 80 for Reaumer's.

The rate of expansion of mercury with rise of temperature increases as the temperature becomes higher; from which it follows, that if a thermometer showing the dilatation of mercury simply were made to agree with an air thermometer at 32° and 211°, the mercurial thermometer would show lower temperatures than the air thermometer between those standard points and higher temperatures beyond them.

In Fahrenheit's time it was supposed that the greatest degree of cold attainable was reached by mixing snow and common salt, or snow and sal-ammoniac. A thermometer plunged into a mixture of this kind was found to fall much below the point indicated by melting ice. The point to which the mercury fell by contraction, when plunged in this mixture, Fahrenheit marked 0, the interval between this and the freezing-point he divided into thirty-two equal divisions, hence the freezing-point came to be indicated by 32°.

Then equal divisions were continued upwards, and the mercury, by expansion, reaching 212,° when the thermometer was immersed in boiling water, this 212° was called the boiling-point. This is briefly the reason for Fahrenheit adopting his method of division, and why he has 212° — 32° = 180° between the freezing- and the boiling-points.

But a much lower temperature than Fahrenheit's 0° has been observed in cold countries, and as mercury becomes solid at 40° Fah. below freezing, it would be the most accurate limit to the scale, as it would register the utmost extremes of heat and cold to which the mercurial thermometer is sensible.

Centigrade Thermometer. — On the scale of this thermometer the space between the freezing- and the boil-

ing-points of water is divided into 100 equal parts, the zero point being placed at freezing. This division being in harmony with our decimal arithmetic, is better adapted than Fahrenheit's or Reaumer's scale for scientific purposes.

Reaumer's Thermometer.—In Reaumer's thermometer the melting-point of ice is taken as zero, and the distance between that and the boiling-point for water is divided into 80 equal parts. Reaumer having observed that between those temperatures spirits of wine (which he used for the thermometric fluid) expanded from 1000 to 1080 parts. This division soon became general in France and other countries, and a great number of valuable observations have been recorded in terms of it; but it is now seldom used in works of science.

Change of Zero.—There is a circumstance connected with the mercurial thermometer which requires to be attended to, when very exact determinations of temperature are to be made, as it has been observed that when thermometers which have been constructed for several years are placed in melting ice, the mercury stands in general higher than the zero point of the scale; and this circumstance, which renders the scale inaccurate, has been usually ascribed to the slowness with which the glass of the bulb acquires its permanent arrangement, after having been heated to a high degree in boiling the mercury.

In very nice experiments it is always necessary to verify the zero point; for it was found that when thermometers have been kept during a certain time in a low temperature, the zero point rises, but falls when they have been kept in a high temperature, and this remark applies equally to old thermometers and to those which have been recently constructed.

Absolute Zero.—Absolute zero is a temperature which

is fixed by reasoning, although no opportunity ever occurs for observing it. It is the temperature corresponding to the disappearance of gaseous elasticity, or, in other words, that point at which gas would become a solid, as when water becomes ice. This temperature is called, in reference to all gases, the absolute zero. The positions of the absolute zero on the ordinary scales would be

On Reaumur's scale	219·2° below 0°.		
On Centigrade scale	274°	"	"
On Fahrenheit's scale	461·2°	"	"

Register Thermometers.—In meteorological observations, it is of great importance to ascertain the limits of the range of the thermometer in a given period of time, or during the absence of the observant. Numerous contrivances have accordingly been proposed for this purpose; but Six's is the most frequently used. It consists of a long cylindrical bulb, with a tube bent in the form of a siphon and terminating in a small cavity; a part of the tube is filled with mercury, but the bulb and the remaining portion of the tube and the cavity are filled with highly rectified alcohol.

The use of the mercury in the middle of the tube is to give motion to two indices, which consist each of a small glass tube in which a small bit of iron-wire is enclosed, the end being capped with enamel.

The indices are of such a size that they freely move within the tube and allow the spirits to pass, but a slender spring attached to each presses against the side of the tube, and prevents the index from falling down when it has been raised to any point, and the mercury recedes.

But this instrument has all the defects which belong to the spirit thermometer, and the indications are in some degree deranged by the expansion and contraction of the

enclosed column of mercury, and, probably, also by the friction of the indices.

Spirit Thermometers are used to measure temperatures at and below the freezing-point of mercury. Their deviations from the air thermometer are greater than those of the mercurial thermometer.

Solid Thermometers. — Solid thermometers are sometimes used, which indicate temperatures by showing the difference between the expansions of a pair of bars of two substances whose rates of expansion are different. When such thermometers are used to indicate temperatures higher than the boiling-point of mercury under one atmosphere (about 676° Fah.), they are called Pyrometers.

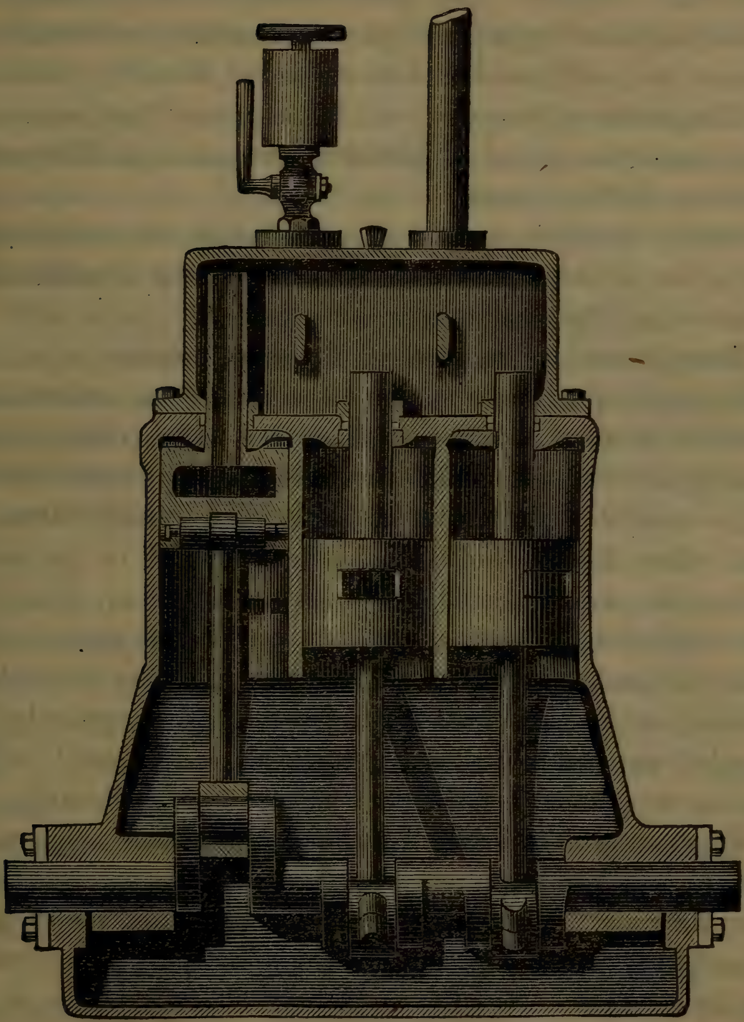
Fixed Temperatures are the boiling-point for water and the melting-point for ice.

Underground Temperatures. — The following table shows the increase of temperature in degrees Fahrenheit, below the surface of the earth, corresponding to depth, etc.

Feet.	Deg. Fah.	Feet.	Deg. Fah.
68	47·9	1,290	58·3
299	48·8	1,414	59·4
621	50·7	1,652	61·2
939	57·8	1,900	61·4

Rules *for comparing Degrees of Temperature indicated by different Thermometers.* — **Rule I.** — Multiply degrees of Centigrade by 9, and divide by 5; or multiply degrees of Reaumur by 9, and divide by 4. Add 32 to the quotient in either case, and the sum is degrees of Fahrenheit.

Rule II. — From degrees of Fahrenheit subtract 32; multiply the remainder by 5, and divide by 9 for degrees of Centigrade; or multiply by 4, and divide by 9 for degrees of Reaumur.



WILLIAMS' THREE-CYLINDER ENGINE.

ELASTIC FLUIDS.

Elastic fluids are divided into two classes — permanent gases and vapors. The gases cannot be converted into the liquid state by any known process of art; whereas the vapors are readily reduced to the liquid form by pressure or diminution of temperature. In respect of their mechanical properties there is, however, no essential difference between the two classes.

Elastic fluids, in a state of equilibrium, are subject to the action of two forces; namely, gravity, and a molecular force acting from particle to particle.

Gravity acts on the gases in the same manner as on all other material substances; but the action of the molecular forces is altogether different from that which takes place among the elementary particles of solids and liquids; for, in the case of solid bodies, the molecules strongly attract each other (hence results their cohesion), and, in the case of liquids, exert a feeble or evanescent attraction, so as to be indifferent to internal motion; but, in the case of the gases, the molecular forces are repulsive, and the molecules yielding to the action of these forces, tend incessantly to recede from each other, and, in fact, do recede until their further separation is prevented by an exterior obstacle.

Thus, air confined within a close vessel exerts a constant pressure against the interior surface, which is not sensible, only because it is balanced by the equal pressure of the atmosphere on the exterior surface. This pressure exerted by the air against the sides of a vessel within which it is confined, is called its elasticity, elastic force, or tension.

Conditions of Equilibrium.— In order that all the parts of an elastic fluid may be in equilibrium, one condition only is necessary; namely, that the elastic force be the same at every point situated in the same horizontal plane.

This condition is likewise necessary to the equilibrium of liquids, and the same circumstances give rise to it in both cases; namely, the mobility of the particles, and the action of gravity upon them.

The density of bodies being inversely as their volumes, the law of Mariotte may be otherwise expressed by saying the density of an elastic fluid is directly proportional to the pressure it sustains. Under the pressure of a single atmosphere, the density of air is about the 770th part of that of water; whence it follows that, under the pressure of 770 atmospheres, air is as dense as water.

The average atmospheric pressure being thus equal to that of a column of water of about 32 feet in altitude at the bottom of the sea, at a depth of 24,640 (equals 770 multiplied by 32) feet, or $4\frac{2}{3}$ miles, air would be heavier than water; and though it should still remain in a gaseous state, it would be incapable of rising to the surface.

CALORIC.

The ordinary application of the word *heat* implies the sensation experienced on touching a body hotter, or of a higher temperature; whilst the term *caloric* provides for the expression of every conceivable existence of temperature.

Caloric is usually treated as if it were a material substance; but, like light and electricity, its true nature has yet to be determined.

Caloric passes through different bodies with different degrees of velocity. This has led to the division of bodies into *conductors* and *non-conductors* of caloric; the former includes such bodies as metals, which allow caloric to pass freely through their substance, and the latter comprises those that do not give an easy passage to it, such as stones, glass, wood, charcoal, etc.

Radiation of Caloric. — When heated bodies are exposed to the air, they lose portions of their heat by projections in right lines into space from all parts of their surface. Radiation is effected by the nature of the surface of the body; thus, black and rough surfaces radiate and absorb more heat than light and polished surfaces. Bodies which radiate heat best, absorb it best.

Reflection of caloric differs from radiation, as the caloric is in this case reflected from the surface without entering the substance of the body. Hence, the body which radiates, and consequently absorbs most caloric, reflects the least, and *vice versa.*

Latent caloric is that which is insensible to the touch, or incapable of being detected by the thermometer. The quantity of heat necessary to enable ice to assume the fluid state, is equal to that which would raise the temperature of the same weight of water 140° Fah., and an equal quantity of heat is set free from water when it assumes the solid form.

Sensible caloric is free and uncombined, passing from one substance to another, affecting the senses, in its passage, determining the height of the thermometer, and giving rise to all the results which are attributed to this active principle.

Evaporation produces cold, because caloric must be absorbed in the formation of vapor, a large quantity of it passing from a sensible to a latent state, the capacity for heat of the vapor formed being greater than that of the fluid from which it proceeds.

Caloric is, therefore, either free and sensible, or latent and insensible. Caloric is known to be the cause of fluidity; and the absence of caloric, the cause of solidity. If heat be applied to ice or iron, it becomes fluid; if exposed to cold, it resumes its solid form.

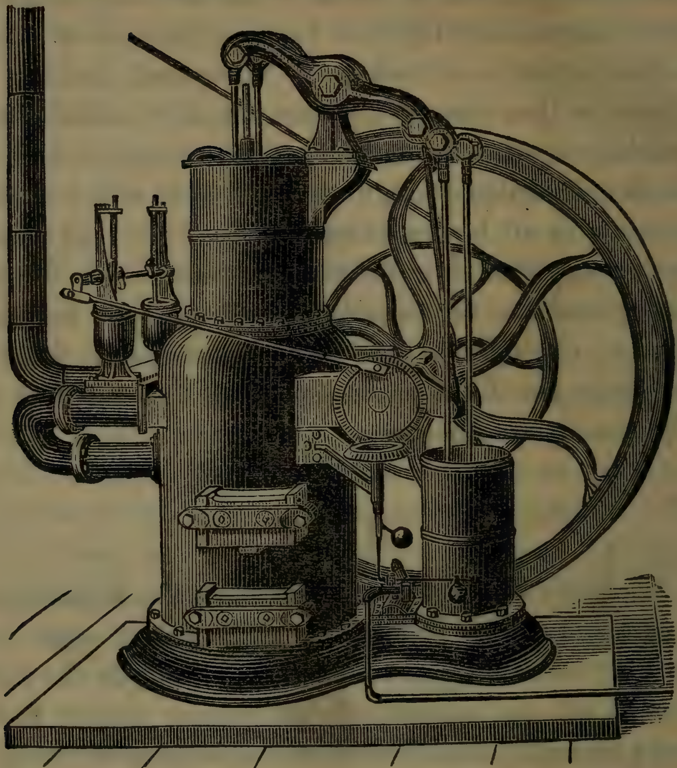
The theory is, that all solid bodies are composed of particles of matter held together by the attraction of cohesion; but that a portion of caloric is interposed between these particles, so that they do not ever actually touch — though appearing to; that when more caloric is applied, the particles are separated by it, or to receive it, which causes an expansion, as all bodies do expand by heat; that when, by a further application of caloric, the particles, to receive it, have separated so far asunder as sufficiently to weaken the attraction of cohesion, the body ceases to be solid, and passes into the fluid state, in which the particles move freely among one another.

HEAT.

It would be difficult to over-estimate the importance of the part played by heat, both on a grand scale in the laboratory of nature, and on a minor scale, in the domain of human art and science. In the former respect, it is not only an essential condition to the existence of life on this planet, but also the prime agent in putting in motion most of the physical changes which take place at the earth's surface.

In the latter, it must be regarded not only as furnishing man with the chief means he possesses of imitating nature, and moulding and modifying natural productions to his wants; but also as bestowing on him the ability to generate and apply at pleasure a force equally stupendous and easy to control.

Heat is one form of mechanical power, or, more properly, a given quantity of heat is the equivalent of a determinate amount of mechanical power; and as heat is capable of producing power, so, contrariwise, power is capable of producing heat.

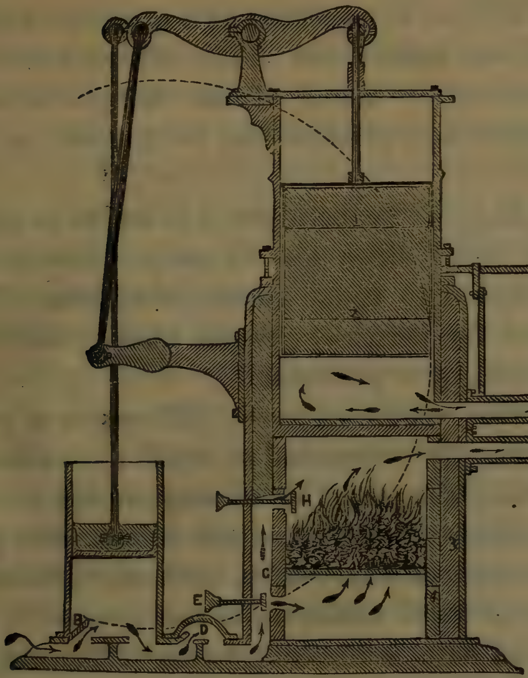


Roper's Caloric Engine.

As it becomes necessary to have a standard for measuring the amount of heat absorbed or evolved during any operation, in this country the standard unit is the amount of heat necessary to raise the temperature of a pound of water 1° Fah., or from 32° to 33° Fah.

Specific Heat.—Different bodies require very different quantities of heat to effect in them the same change of temperature. The capacity of a body for heat is termed its "specific heat," and may be defined as the number of units of heat necessary to raise the temperature of 1 pound of that body 1° Fah.

When a substance is heated it expands, and its temperature is increased. It is evident, therefore, that heat is



Section of Roper's Caloric Engine.

required both to raise the temperature and to increase the distance between the particles of the substance.

The heat used in the latter case is converted into interior work, and is not sensible to the thermometer; but it will be given out, if the temperature of the substance is reduced to the original point.

Thus, while heat is apparently lost, it is only stored up, ready to do work, and the substance possesses a certain amount of potential energy, or possibility of doing work.

Now, as different substances vary greatly in their molecular constitution, expanding and contracting the same amount with widely differing degrees of force, it is to be expected that the quantity of heat that will raise one substance to a given temperature may produce a less or greater degree of sensible heat to another; and we find in practice that such is the case.

The condition of heat is measured as a quantity, and its amounts in different bodies and under different circumstances are compared by means of the changes in some measurable phenomenon produced by its transfer or disappearance.

In so using changes of temperature, it is not to be taken for granted that equal differences of temperature in the same body correspond to equal quantities of heat. This is the case, indeed, for perfectly gaseous bodies; but that is a fact only known by experiment.

On bodies in other conditions, equal differences of temperature do not exactly correspond to equal quantities of heat. To ascertain, therefore, by an experiment on the changes of temperature of any given substances, what proportion two quantities of heat bear to each other, the only method which is of itself sufficient, in the absence of all other experimental data, is the comparison of the weights of that substance which are raised from the same low temperature to a high or fixed temperature.

The Unit of Heat.—The unit of heat, or thermal unit employed, is the quantity of heat, as before stated, that would raise 1 pound of pure water 1° Fah., or from 39° to 40° Fah.

The reason for selecting that part of the scale which is nearest the temperature of the greatest density of water, is because the quantity of heat corresponding to an interval of one degree in a given weight of water is not exactly the same in different parts of the scale of temperature.

Latent Heat.—Latent heat means a quantity of heat which has disappeared, having been employed to produce some change other than elevation of temperature. By exactly reversing that change, the quantity of heat which had disappeared is reproduced.

When a body is said to possess or contain so much latent heat, what is meant is simply this: that the body is

in a condition into which it was brought from a former different condition by transferring to it a quantity of heat which did not raise its temperature, the change of condition having been different from change of temperature, and that by restoring the body to its original condition in such a manner as exactly to reverse the former process, the quantity of heat formerly expended can be reproduced in the body and transferred to other bodies.

When a body passes from the solid to the liquid state, its temperature remains stationary, or nearly so, at a certain melting point, during the whole operation of melting, and in order to make that operation go on, a quantity of heat must be transferred to the substance melted, having a certain amount for each unit of weight of the substance. That heat does not raise the temperature of the substance, but disappears in causing its condition to change from the solid to the liquid state.

When a substance passes from the liquid to the solid state, its temperature remains stationary, or nearly so, during the whole operation of freezing; a quantity of heat equal to the latent heat of fusion is produced in the body, and in order that the operation of freezing may go on, that heat must be transferred from that body to some other substance.

Sensible Heat.—Sensible heat is that which is sensible to the touch or measurable by the thermometer.

Mechanical Equivalent of Heat.—The mechanical equivalent of heat is the amount of work performed by the conversion of one unit of heat into work. This has been determined to be equal in amount to the work required to raise 772 pounds one foot high, or one pound 772 feet high. And as heat and work are mutually convertible, if a body weighing one pound, after falling through a height of 772 feet, were to have its motion suddenly arrested, it

would develop sufficient heat to raise the temperature of a pound of water one degree.

If a pound of water, at a temperature of 212° Fah., is converted into steam, the latter will have a volume of about 27¼ cubic feet. Now, suppose that the water is evaporated in a long cylinder of exactly one foot cross section, open to the atmosphere at the top. When all the water in the cylinder has disappeared, there will be a column of steam 27¼ feet high, which has risen to this height against the pressure of the atmosphere.

The pressure of the air being nearly 15 pounds per square inch, the pressure per square foot is 2115 pounds; and the external work performed by the water, in changing into steam, will be an amount required to raise 2115 pounds to a height of 27¼ feet, or about 57,644 foot-pounds.

Now, since 772 foot-pounds of work require one unit of heat, the external work will take up 57,644 divided by 772, equals 74·64 units of heat.

But it has been shown that the total number of units of heat required to change water into steam is about 968 (more accurately, 966·6). Hence the internal work will be equal to an amount developed by the conversion of 966·6 less 74·67, equals 891·93 units of heat into work, and this will equal 891·93, multiplied by 772 equals 688,569 foot-pounds.

Mechanical Theory of Heat.—The mechanical theory of heat is now generally adopted. It considers that heat and work are interchangeable, and on this theory can be explained what becomes of the latent heat. All solid bodies are supposed to be made up of molecules, which are not in contact, but are prevented from separating by a force called cohesion.

If a body is heated to a sufficient temperature, the force

of expansion becomes equal to that of cohesion, and the body is liquefied; and if still more heat is applied, the force of expansion exceeds that of cohesion, and the liquid becomes a vapor.

But in each of these changes work is performed, and the heat that is supplied is converted into work.

For instance, if ice is at a temperature of 32°, and heat is applied, this is converted into the work that is developed in changing the ice into water, and we say that heat becomes latent, and when water is at 212°, and we continue to apply heat; this is converted into the work that must be done in changing the water into steam.

Dynamic Equivalent of Heat.—It is a matter of ordinary observation that heat, by expanding bodies, is a source of mechanical energy; and conversely, that mechanical energy, being expended either in compressing bodies or in friction, is a source of heat.

In all other cases in which heat is produced by the expenditure of mechanical energy, or mechanical energy by the expenditure of heat, some other change is produced besides that which is principally considered; and this prevents the heat and the mechanical energy from being exactly equivalent.

Power of Expansion by Heat.—When bodies expand, the molecules of which they are composed are pushed farther asunder by the oscillatory motion communicated to them. The heat may be described as entering the substance and immediately setting to work to separate the particles. The power or energy it exerts to do this is immense.

Molecular or Atomic Force of Heat.—All molecules are under the influence of two opposite forces. The one, molecular attraction, tends to bring them together; the other, heat, tends to separate them; its intensity varies

with its velocity of vibration. Molecular attraction is only exerted at infinitely small distances, and is known under the name of cohesion, affinity, and adhesion.

Total or Actual Heat.—If, when a substance, by the expenditure of energy in friction, is brought from a condition of total privation of heat to any particular condition of heat, we subtract from the total energy so expended, first, the mechanical work performed by the action of the substance on external bodies, through changes of its volume, during such heating; secondly, the mechanical work due to mutual actions between the particles of the substance itself during such heating, the remainder will represent the energy which is employed in making the substance hot.

Communication of Heat.—Heat may be communicated from a hot body to a cold one in three ways,—by radiation, conduction, and circulation.

The rapidity with which heat radiates varies, other things being equal, as the square of the temperature of the hot body in excess of the temperature of the cold one; so that a body, if made twice as hot, will lose a degree of temperature in one-fourth of the time; if made three times as hot, it will lose a degree of temperature in one-ninth of the time, and so on in all other proportions.

Transmission of Heat.—Tredgold and others have made experiments to ascertain the rate at which heat is transferred from metal to gases and from gases to metal. Other things being equal, it has been found that the rate of transference is as the difference of temperature. But in practice the conditions are different from those in the experiment; generally, in experiments, the air has been still, and the gases moving under natural draught; but in locomotive practice, the velocity of the gases is so great as to render the results of most experiments inapplicable.

Effects of Heat on the Circulation of Water in Boilers.—As the particles of water rise heated from the bottom of the boiler, other particles necessarily subside into their places, and it is a point of considerable importance to ascertain the direction in which the currents approach the plate to receive heat. A particle of water cannot leave the heated plate until there is another particle at hand to occupy its position; and, therefore, unless a due succession in the particles is provided for, the plate cannot get rid of its heat, and the proper formation of steam is hindered.

But it must be understood that vaporizing does not depend on the quantity of heat applied to the plate, but on the quantity of heat abstracted from it by the particles of water.

Medium Heat.—The medium heat of the globe is placed at 50°; at the torrid zone, 75°; at moderate climates, 50°; near the Polar regions, 36° Fah.

The extremes of natural heat are from 70° to 120°; of artificial heat, 91° to 36000° Fah.

LATENT HEAT OF VARIOUS SUBSTANCES.

	Fah.		Fah.
Ice	140°	Steam	990°
Sulphur	144	Vinegar	875
Lead	162	Ammonia	860
Beeswax	176	Alcohol	442
Zinc	493	Ether	301

TABLE OF THE RADIATING POWER OF DIFFERENT BODIES.

Water	100	Blackened tin	100
Lamp-black	100	Clean "	12
Writing-paper	100	Scraped "	16
Glass	90	Ice	85
India-ink	88	Mercury	20
Bright lead	19	Polished iron	15
Silver	12	Copper	12

TABLE

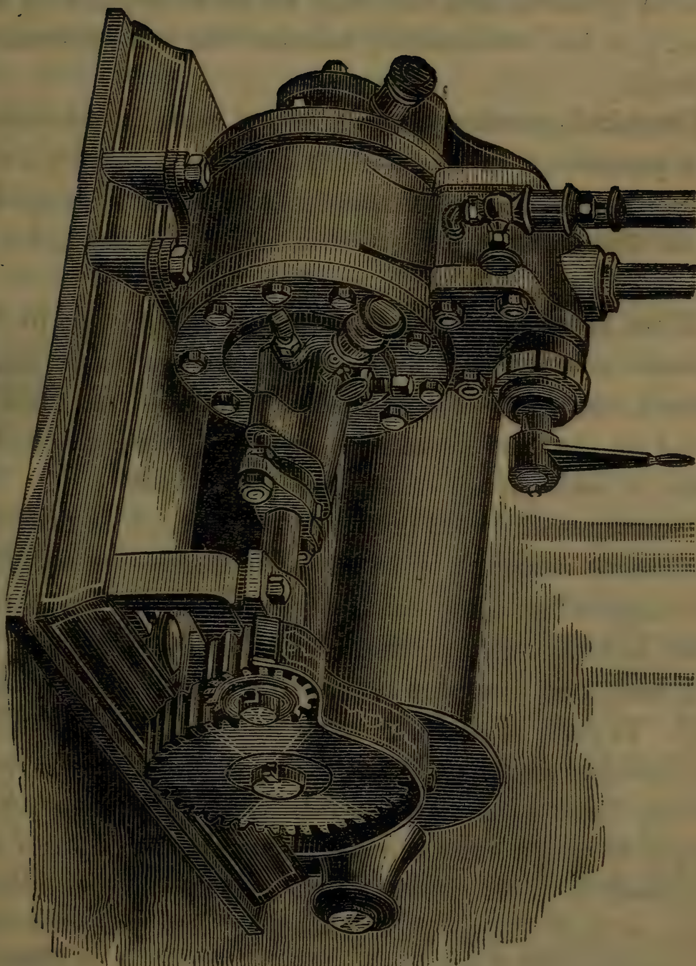
SHOWING THE EFFECTS OF HEAT UPON DIFFERENT BODIES.

	Fah.		Fah.
Cast-iron, thoroughly smelted	2754°	Lead melts	594°
Fine gold melts	1983	Bismuth "	476
Fine silver "	1850	Tin "	421
Copper "	2160	Tin and Bismuth, equal parts, melt	283
Brass "	1900	Tin, 3 parts, Bismuth 5, and Lead 2 parts, melt	212
Red heat, visible by day	1077	Alcohol boils	174
Iron red-hot in twilight	884	Ether "	98
Common fire	790	Human blood (heat of)	98
Iron, bright red in the dark	752	Strong wine freezes	20
Zinc melts	740	Brandy "	7
Quicksilver boils	630	Mercury melts	39
Linseed oil	600		

COMBUSTION.

Combustion is, strictly speaking, the development of heat by chemical combination; but though this may take place from the union of a variety of bodies, the omnipresent agent, oxygen, plays so vastly more important a part than all others in the disengagement of light and heat, that the act of its combination with other bodies is pre-eminently entitled combustion.

Since combustion, in the ordinary acceptation of the word, is the only means had recourse to in the arts for the development of artificial heat, perfect combustion may, for our purpose, be defined to be — the combination of a combustible body with the largest measure of oxygen with which it is capable of uniting. In fact, for all practical purposes, the fuel, or combustible body, employed may be regarded as composed exclusively of carbon and hydrogen; so that our inquiry becomes narrowed to the combinations of oxygen with these two elementary substances.



MASSEY'S ROTARY ENGINE.

No substance in nature is combustible of itself, to whatever degree of heat it may be exposed; nor can it be ignited only when in presence of or in mechanical mixture with air, or its vital element, oxygen, because combustion is continuous ignition, and can only be made to exist by maintaining in the combustible mixture the heat necessary to ignite it.

Chemical combination, in every case, is accompanied by a production of heat; every decomposition, by a disappearance of heat equal in amount to that which is produced by the combination of the elements which are to be separated.

When a complex chemical action takes place in which various combinations and decompositions occur simultaneously, the heat obtained is the excess of the heat produced by the combinations above the heat, which disappears in consequence of the decompositions.

Sometimes the heat produced is subject to a further deduction, on account of heat which disappears in melting or evaporating some of the substances which combine either before or during the act of combination.

Substances combine chemically in certain proportions only. To each of the substances known in chemistry, a certain number can be assigned, called its chemical equivalent, having these properties: — 1st. That the proportions by weight in which substances combine chemically can all be expressed by their chemical equivalents, or by simple multiples of their chemical equivalents. 2d. That the chemical equivalent of a compound is the sum of the chemical equivalents of its constituents.

Chemical equivalents are sometimes called atomic weights or atoms, in accordance with the hypothesis that they are proportionate to the weights of the supposed atoms of bodies, or smallest similar parts into which bodies

are assumed to be divisible by known forces. The term *atom* is convenient from its shortness, and can be used to mean "chemical equivalent," without necessarily affirming or denying the hypothesis from which it is derived, and which, how probable soever it may be, is, like other molecular hypotheses, incapable of absolute proof.

The chief elementary combustible constituents of ordinary fuel are carbon and hydrogen. Sulphur is another combustible constituent of ordinary fuel, but its quantity is small and its heating power of no practical value.

Coal is composed, so far as combustion is concerned, of solid carbon and a gas consisting of hydrogen and carbon.

When the coal is heated, it first discharges its gas; the solid carbon left then ignites in presence of oxygen, and will retain the temperature necessary to combustion so long as oxygen is applied.

The Ingredients of Fuel.—Fixed or free carbon which is left in the form of charcoal or coke after the volatile ingredients of the fuel have been distilled away. This ingredient burns either wholly in the solid or partly in the solid or gaseous state; the latter part being first dissolved by previously formed carbonic acid, as already explained.

Hydrocarbons, such as gas, pitch, tar, naphtha, etc., all of which must pass into the gaseous state before being burned. If mixed on their first issuing from among the burning carbon with a large quantity of air, these inflammable gases are completely burned, with a transparent blue flame, producing carbonic acid and steam.

Mixture of Fuel and Air.—In burning charcoal, coke, and coals with a small proportion only of hydrocarbons, a supply of air sufficient for complete combustion will enter from the ash-pit through the bars of the grate, provided there is a sufficient draught, and that care is taken to distribute the fresh fuel evenly over the fire, and in moderate quantities at a time.

Available Heat of Combustion. — The available heat of combustion of one pound of a given sort of fuel is that part of the total heat of combustion which is communicated to the body, to heat which the fuel is burned.

Anthracite Coal. — The chemical composition of anthracite coal is similar to charcoal, from which it differs chiefly in its form, being very hard and compact, and in the greater quantity of ashes which it contains. It is, like charcoal, unaltered in form after exposure to the strongest heat; even after passing through a blast furnace it has equally as sharp edges, and is in form exactly as it was before.

COMPOSITION OF DIFFERENT KINDS OF ANTHRACITE COAL.

	Carbon.	Volatile matter.	Ashes.	Specific gravity.
Lehigh Coal	88.50	7.50	4.00	......
Schuylkill Coal	92.07	5.03	2.90	1.57
Pottsville	94.10	1.40	4.50	1.50
Pinegrove	79.57	7.15	3.28	1.54
Wilkesbarre	88.90	7.68	3.49	1.40
Carbondale	90.23	7.07	2.70	1.40

The analysis of anthracite shows good coal of that class to be composed of 90·45 carbon, 2·43 hydrogen, 2·45 oxygen, some nitrogen, and 4·67 ashes.

The ashes generally consist, like those of bituminous coal, of silex, alumina, oxide of iron, and chlorides, which generally evaporate and condense on cold objects in the form of white films.

Anthracite is not so inflammable as either dry wood or bituminous coal, but it may be made to burn quite as vividly as either, by exposing it to a strong draught, or in a large mass to the action of the air.

The Quantity of Air required for the Combustion of Anthracite Coal. — In view of the quantity of oxygen required to unite chemically with the various constituents of the coal, we find that in 100 pounds of anthracite coal, consisting of 91 per cent. of carbon and 9 per cent. of the other matter, it will be necessary to have 243·84 pounds of oxygen, since to saturate a pound of carbon by the formation of carbonic acid requires $2\frac{2}{3}$ pounds of oxygen. To saturate a pound of hydrogen in the formation of water requires 8 pounds of oxygen; hence 3·46 pounds of hydrogen will take 27·68 pounds of oxygen for its saturation.

If then we add 243·84 pounds of oxygen for its saturation, 271·52 pounds of oxygen are required for the combustion of 100 pounds of coal.

A given weight of air contains nearly 23·32 per cent. of oxygen; hence to obtain 271·52 pounds of oxygen, we must have about four times that quantity of atmospheric air, or, more accurately, 1164 pounds of air for the combustion of 100 pounds of coal.

A cubic foot of air at ordinary temperatures weighs about ·075 pound; so that 100 pounds of coal require 15,524 cubic feet of air, or one pound of coal requires about 155 cubic feet of air, supposing every atom of the oxygen to enter into combination.

If, then, from one-third to one-half of the air passes unconsumed though the fire, an allowance of 240 cubic feet of air for each pound of coal will be a small enough allowance to answer the requirements of practice, and in some cases as much as 320 cubic feet will be required.

The Evaporative Efficiency of a Pound of Anthracite Coal. — The evaporative efficiency of a pound of carbon has been found, experimentally, to be equivalent to that necessary to raise 14,000 pounds of water through one degree, or 14 pounds of water through 1000 degrees, sup-

posing the whole heat generated to be absorbed by the water.

Now, if the water be raised into steam from a temperature of 60°, then 1118·9° of heat will have to be imparted to it to convert it into steam of 15 pounds pressure per square inch; 14,000 divided by 1118·9 equals 12·5 pounds will be the number of pounds of water, therefore, which a pound of carbon can raise into steam of 15 pounds pressure from a temperature of 60°. This, however, is a considerably larger result than can be expected in practice.

Bituminous Coal.—Under this class we range all that mineral coal which forms coke, that is, it swells upon being exposed to heat, burns with a bright flame, blazes, and after the flame disappears there remains a spongy, porous mass — coke — which burns without flame like charcoal.

In its composition we find chiefly carbon, oxygen, hydrogen, nitrogen, sulphur, and ashes, with a little water, which has been absorbed by the crevices.

The following table shows the comparative composition of various sorts of mineral fuel:—

	Carbon.	Hydrogen.	Oxygen and Nitrogen.	Ashes.
Turf	58.09	5.93	31.37	4.61
Brown Coal	71.71	4.85	21.67	1.77
Hard Bituminous Coal	82.92	6.49	10.86	0.13
Cannel Coal	83.75	5.66	8.04	2.55
Cooking or Baking Coal	87.95	5.24	5.41	1.40
Anthracite	91.98	3.92	3.16	0.94

An essential condition in forming coke is that the coal, on being heated, swells and changes into irregular, spongy masses, which adhere intimately together. This operation is designed to expel sulphur and hydrogen, and form a

coal which is not altered by heat. The sulphur cannot be entirely separated from coke, or from carbon, no matter how high the heat may be; neither can all the hydrogen be removed from carbon by simply heating the compound. If oxygen is admitted to these combinations, both sulphur and hydrogen may be almost entirely expelled, that is, provided the oxygen is not introduced under too high or too low a heat.

The most important point, and one which has a direct bearing upon the value of coal, is the quantity of heat which it can evolve in combustion.

If we assume that the quantity of ashes is equal in the four substances mentioned below, that is, 5 per cent. in each, and suppose further that pine charcoal furnishes 100 parts of heat, the following table shows the quantity which must be liberated in their perfect combustion:

Kind of Coal.	Carbon.	Hydrogen.	Water.	Quantity of Heat.
Brown Coal......................	69	3	23	78
Cooking Coal....................	75	4	16	87
" "	78	4	13	90
Anthracite Coal..................	85	3	7	94
Pure Carbon......................	100	...	...	100

Bituminous coal, like all other fuel, is a compound substance, which may be decomposed by heat into several distinct elements—generally five or six, at least. So far as relates to combustion, we are concerned principally with but two of these, viz., solid carbon, represented by coke, and hydrogen, generally known under the indefinite term of "gas." These two elements contain principally the full heating qualities of the coal. The carbon, so long as it remains as such, is always solid and visible.

The hydrogen, when driven from the coal by heat,

carries with it a portion of carbon, the gaseous compound being known as carburetted hydrogen.

A ton of 2000 pounds of average bituminous coal contains, say 1600 pounds, or 80 per cent. of carbon, 100 pounds, or 5 per cent. of hydrogen, and 300 pounds, or 15 per. cent. of oxygen, nitrogen, sulphur, sand, and ashes.

But if this coal be coked, the 100 pounds of hydrogen driven off by heat will carry about 300 pounds of carbon in combination with it, making 400 pounds, or nearly 1000 cubic feet of carburetted hydrogen gas.

But still 1300 pounds of carbon (65 per cent. of the original coal) will be left, and, with the earthy matter, ashes, sulphur, etc., retained with it, the coke will weigh but about 1350 or 1400 pounds,—67½ to 70 per cent. of the original coal.

The only proportions in which carbon and hydrogen combine with air in combustion are these:

For every pound of carbon (pure coke), 12 pounds (equal to 159½ cubic feet) of air are required to combine intimately with it.

For every pound of hydrogen, 36 pounds (equal to 478½ cubic feet) of air are required to be similarly combined.

Thus for every pound of carburetted hydrogen gas, being one-fourth pound of hydrogen and three-fourths of a pound of carbon, 18 pounds (equal to 239¼ cubic feet) of air are required to be combined with it.

These are the elements and their combining proportions that have to be dealt with in a locomotive furnace. For every 2000 pounds of coal burned, the 400 pounds of carburetted hydrogen—the "gas"—require 95,700 cubic feet of atmospheric air at ordinary temperature, and the 1300 pounds of solid carbon require 207,350 cubic feet of air. Practically, the gas from a ton of ordinary bitu-

minous coal requires 100,000 cubic feet of air for its combustion, while the remaining coke requires 200,000 feet. Thus the gaseous matter of the coal requires one-half as much air as is taken up by the solid coke.

The heating value of any combustibles is exactly proportional to the quantity of air with which it will combine in combustion. Hence hydrogen, which combines with three times the quantity of air (oxygen) which would be taken up by carbon, has, for equal weights, three times the heating value. Thus, the 100 pounds of pure hydrogen in a ton of coal have the same heating efficiency as that due to 300 pounds of the remaining carbon or pure coke.

It will now be seen that complete combustion cannot produce smoke, since smoke contains a quantity of unburnt matter, and is in itself a proof of incomplete combustion. The products of perfect combustion are invisible — being for carbon and oxygen, carbonic acid; and for hydrogen and oxygen, invisible steam, which condenses into water.

The admission of heated air to furnaces or fire-boxes of locomotives can be of no practical value, since for every 493° Fah. of heat added, its original bulk or volume is doubled; treble at 1010° Fah.; so that at 3000° Fah. the heated air in the interior of the furnace has six times its original volume. This makes it more unmanageable, and as its contained oxygen remains the same in weight, its mixture with the gas becomes more difficult, while, when mixed, it can do only the same work as before.

Waste of Unburnt Fuel.—This generally arises from the brittleness of the fuel, combined with want of care on the part of the fireman, by which cause the fuel is made to fall into small pieces, which escape between the grate-bars into the ash-pit, and are lost.

It is almost impossible to estimate the loss of fuel occa-

sioned by carelessness and bad firing, but the amount which is unavoidable, even with care and good firing, has been ascertained by experiment to range from 2½ to 3 per cent. of the fuel consumed.

TABLE
SHOWING THE TOTAL HEAT OF COMBUSTION OF VARIOUS FUELS.

SORT OF FUEL.	Equivalent in pure Carbon.	Evaporative power in lbs. water from 212° Fah.	Total heat of combustion in lbs. water heated 1° Fah.
Charcoal	0.93	14.00	13500
Charred Peat	0.80	12.00	11600
Coke—good	0.94	14.00	13620
" mean	0.88	13.20	12760
" bad	0.82	12.30	11890
COAL.—Anthracite	1.05	15.75	15225
Hard Bituminous—hardest	1 06	15.90	15370
" " softest	0.95	14.25	13775
Cooking coal	1.07	16.00	15837
Canning coal	1.04	15.60	15080
Long flaming splint coal	0.91	13.65	13195
Lignite	0.81	12.15	11745
PEAT.—Perfectly air-dry	0.66	10.00	9660
Containing 25 per ct. water	......	7.25	7000
WOOD.—Perfectly air-dry	0.50	7.50	7245
Containing 20 per ct. water	......	5.80	5600

Spontaneous Combustion.—A great deal has been said and written on the subject of spontaneous combustion, and the danger likely to result from allowing steam-pipes to come in contact with the wood-work in buildings; but as the temperature of superheated steam ranges from 300° to 500° Fah., it is only able to set fire to such substances as sulphur, gun-cotton, and nitro-glycerine. It is, perhaps, able to fire gunpowder, but certainly cannot ignite wood.

It is only when dried wood, sawdust, or rags have been saturated by drying oil or other equivalents, that the temperature may be indefinitely raised, and finally reach 400° or 500° Fah., or until the point of inflammability is attained. This is caused by the oxidation of the oil and the agency of the air.

TABLE

SHOWING THE NATURE AND VALUE OF SEVERAL VARIETIES OF AMERICAN COAL AND COKE, AS DEDUCED FROM EXPERIMENTS BY PROFESSOR JOHNSON, FOR THE UNITED STATES GOVERNMENT.

Designation of Fuel.	Specific gravity.	Weight per cubic foot.	Lbs. of steam from water at 212° by 1 lb. of fuel.	Lbs. of steam from water at 212° by 1 cub.ft.of fuel.	Weight of clinker from 100lbs.of fuel.	No. of cub. ft. required to stow a ton.
BITUMINOUS.						
Cumberland, *maximum*	1.313	52.92	10.7	573.3	2.13	42.3
" *minimum*	1.337	54.29	9.44	532.3	4.53	41.2
Blossburgh	1.324	53.05	9.72	522.6	3.40	42.2
Midlothian, *screened*..	1.283	45.72	8.94	438.4	3.33	49.0
" *average*..	1.294	54.04	8.29	461.6	8.82	41.4
Newcastle	1.257	50.82	8.66	453.9	3.14	44.0
Pictou	1.318	49.25	8.41	478.7	6.13	45.0
Pittsburgh	1.252	46.81	8.20	384.1	.94	47.8
Sydney	1.338	47.44	7.99	386.1	2.25	47.2
Liverpool	1.262	47.88	7.84	411.2	1.86	46.7
Clover Hill	1.285	45.49	7.67	359.3	3.86	49.2
Cannelton, Ia	1.273	47.65	7.34	360.0	1.64	47.0
Scotch	1.519	51.09	6.95	369.1	5.63	43.8
ANTHRACITE.						
Peach Mountain........	1.464	53.79	10.11	581.3	3.03	41.6
Forest Improvement...	1.477	53.66	10.06	577.3	.81	41.7
Beaver Meadow No. 5..	1.554	56.19	9.88	572.9	.60	39.8
Lackawanna	1.421	48.89	9.79	493.0	1.24	45.8
Beaver Meadow No. 3.	1.610	54.93	9.21	526.5	1.01	40.7
Lehigh	1.590	55.32	8.93	515.4	1.08	40.5
COKE.*						
Natural Virginia.......	1.323	46.64	8.47	407.9	5.31	48·3
Midlothian	....	32.70	8.63	282.5	10.51	68.5
Cumberland	....	31.57	8.99	284.0	3.55	70.9

The fuel value of wood, as compared with coal, is about as follows:

1 Cord air-dried Hickory, or Hard Maple	= 2000 lbs. coal.
1 Cord air-dried White Oak	= 1725 " "
1 Cord air-dried Beech, Red Oak, or Black Oak	= 1450 " "
1 Cord air-dried Poplar, Chestnut, or Elm	= 1050 " "
1 Cord air-dried Average of Pine Wood	= 925 " "

* See page 338.

TABLE

SHOWING SOME OF THE PROMINENT QUALITIES IN THE PRINCIPAL AMERICAN WOODS.

Species.	Specific gravity, green.	Specific gravity, air-dried.	Specific gravity, kiln-dried.	Degrees of heat which may be generated.	Percentage of charcoal.	Quantity of heat as to volume.	Weight of one cord in pounds.	Relative value as fuel.
Hickory......	....	....	....	3000	44.69	25	4496	1.00
White Oak..	1.07	0.71	0.66	3000	21.62	25	3821	0.81
Black Oak...	....	....	....	3000	23.80	25	3254	0.71
Red Oak.....	1.05	0.68	0.66	3000	22.43	25	3254	0.69
Beech.........	0.98	0.59	0.58	3000	32.36	25	3236	0.65
Birch.........	0.90	0.63	0.57	3000	....	25	....	....
Maple	0.90	0.64	0.61	3000	27.00	25	2700	0.57
Yellow Pine	....	....	....	2800	24.63	23	2463	0.54
Chestnut.....	....	....	....	3000	25.25	25	2333	0.52
Pitch Pine...	....	....	....	2800	19.04	23	1904	0.43
White Pine..	0.87	0,47	0.38	2800	18.68	23	1868	0.42

TABLE

SHOWING THE RELATIVE PROPERTIES OF GOOD COKE, COAL, AND WOOD.

Name of Fuel.	Weight per cubic foot, in pounds.	Degrees of heat generated.	Percentage of carbon in the fuel.	Economical bulk, or cubic feet required to stow one ton.	Economic or stowage weight per cubic foot.	Cubic feet of air to evaporate one pound of water.	Equivalent economic bulk, to evaporate same weight of water.	Weight of water evaporated per pound of fuel in ordinary practice.	Relative value as fuel, disregarding actual cost.
Coke.............	63	4300	95	80	28	22.4	13	8½	100
Coal.............	80	4000	88	44	51	32.0	10	6	71
Wood............	30	2800	20	107	21	16.0	60	2½	29

TABLE

OF TEMPERATURES REQUIRED FOR THE IGNITION OF DIFFERENT COMBUSTIBLE SUBSTANCES.

Substances.	Temperature of Ignition.	Remarks.
Phosphorus	140°	Melts at 110°.
Bisulphide of carbon vapor	300°	Melts at 130°.
Fulminating Powder	374°	Used in percussion caps.
Fulminate of Mercury	392°	According to Legue and Champion.
Equal parts of chlorate of potash and sulphur.	395°	
Sulphur	400°	Melts, 280°; boils, 850°.
Gun-cotton	428°	According to Legue and Champion.
Nitro-glycerine	494°	" " "
Rifle-powder	550°	" " "
Gunpowder, coarse	563°	" " "
Picrate of mercury, lead or iron	565°	" " "
Picrate powder for torpedoes	570°	" " "
Picrate powder for muskets	576°	" " "
Charcoal, the most inflammable willow used for gunpowder	580°	According to Pelouse and Fremy.
Charcoal made by distilling wood at 500°	660°	" " "
Charcoal made at 600°	700°	" " "
Picrate powder for cannon	716°	
Very dry wood, pine	800°	
" " " oak	900°	
Charcoal made at 800°	900°	

It will be seen by the above table that the most combustible substances generally considered very dangerous, will only ignite by heat alone at a high temperature, so that for their prompt ignition it requires the actual contact of a spark.

GASES.

All substances, whether animal, vegetable, or mineral, consisting of carbon, hydrogen, and oxygen, when exposed to a red heat, produce various inflammable elastic fluids, capable of furnishing artificial light. We perceive the evolution of this elastic fluid during the combustion of coal in a common fire.

Bituminous coal, when heated to a certain degree, swells and kindles, and frequently emits remarkably bright streams of flame, and after a certain period these appearances cease, and the coal glows with a red light.

The flame produced from coal, oil, wax, tallow, or other bodies which are composed of carbon and hydrogen, proceeds from the production of carburetted hydrogen gas, evolved from the combustible body when in an ignited state.

If coal, instead of being burnt in the way now stated, is submitted to a temperature of ignition in close vessels, all its immediate constituent parts may be collected. The bituminous part is distilled over in the form of coal-tar, etc., and a large quantity of an aqueous fluid is disengaged at the same time, mixed with a portion of essential oil and various ammoniacal salts.

A large quantity of carburetted hydrogen, carbonic oxide, carbonic acid, and sulphuretted hydrogen also make their appearance, together with small quantities of cyanogen, nitrogen, and free hydrogen; and the fixed base of the coal alone remains behind in the distillatory apparatus in the form of a carbonaceous substance called coke. An analysis of coal is thus effected by the process of destructive distillation.

Hydrogen. — Hydrogen is the lightest of all known gases, its specific gravity being only 0·06896. This gas is colorless, and, when perfectly pure, inodorous. It has a

powerful affinity for oxygen, and is therefore eminently combustible. Intense heat is developed by the combustion of hydrogen in oxygen gas, and but little light.

Carbon. — Carbon is well known under the form of coke, charcoal, lamp-black, etc. It is one of the principal constituents of all varieties of coal, and is the basis of the illuminating gases. It is a colorless and inodorous gas, rather lighter than common air, having a specific gravity of 0·9727, is sparingly absorbed by water, and does not precipitate lime-water. It is inflammable, burning with a beautiful blue flame; the product of its combustion is carbonic acid.

Carbon unites with hydrogen in many proportions, and many of these compounds are produced during the distillation of coal; but the only two of importance are carburetted hydrogen and olefiant gas.

Carburetted Hydrogen. — Carburetted hydrogen is abundantly formed in nature, in stagnant pools, ditches, etc., wherever vegetables are undergoing the process of putrefaction; it also forms the greater part of the gas obtained from coal. Carburetted hydrogen consists of 100 volumes of vapor of carbon, and 200 of hydrogen. It is colorless and almost inodorous; it is not dissolved to any extent by water, and is much lighter than atmospheric air, its density being 0·5594. It is very inflammable, burning with a strong yellow flame. The products of its combustion are carbonic acid and water.

Carburetted hydrogen, or coal-gas, when freed from the obnoxious foreign gases, may be propelled in streams out of small apertures, which, when lighted, form jets of flame, which are called gas-lights.

Olefiant Gas.— Olefiant gas is a product of the distillation of oil, resin, and also of coal, when the process is well conducted. It is colorless, tasteless, and without smell

when pure. Water dissolves about one-eighth of its bulk of this gas. It is formed of two volumes of hydrogen, and two of the vapor of carbon condensed into one volume.

Olefiant gas burns with an intense white light, and requires a larger portion of oxygen for its combustion, one volume of the gas requiring not less than three volumes of pure oxygen, or fifteen volumes of atmospheric air for decomposition. The products of the combustion are water and carbonic acid.

Nitrogen.— Nitrogen is one of the constituents of coal. It has the properties of extinguishing burning bodies, and is not absorbed by water; its specific gravity is 0.9760, being lighter than common air, in which it forms a constituent part.

Liquefaction of Gases.— Many of the gases have already been brought into the liquid state by the conjoint agency of cold and compression, and all of them are probably susceptible of a similar reduction by the use of means sufficiently powerful for the required end.

They must consequently be regarded as the superheated steams or vapors of the liquids into which they are compressed.

Compression and Dilatation of Gases.—When a gas or vapor is compressed into half its original bulk, its pressure is double; when compressed into a third of its original bulk, its pressure is treble; when compressed into a fourth of its original bulk, its pressure is quadrupled; and generally the pressure varies inversely as the bulk into which the gas is compressed.

So in like manner if the volume be doubled, the pressure is made one-half of what it was before — the pressure being in every case reckoned from 0, or from a perfect vacuum.

Thus, if we take the average pressure of the atmosphere

at 14·7 pounds on the square inch, a cubic foot of air, if suffered to expand into twice its bulk by being placed in a vacuum measuring two cubic feet, will have a pressure of 7·35 pounds above a perfect vacuum, and also of 7·35 pounds below the atmospheric pressure; whereas, if the cubic foot be compressed into a space of half a cubic foot, the pressure will become 29·4 pounds above a perfect vacuum, and 14·7 pounds above the atmospheric pressure.

The specific gravity of any one gas to that of another will not exactly conform to the same ratio under different degrees of heat, and other pressures of the atmosphere.

Water, as before stated, is composed of two gases, oxygen and hydrogen—the weight of a cubic foot of hydrogen being ·005592 pounds, and of half a cubic foot of oxygen ·0044628 pounds avoirdupois. One cubic foot of hydrogen and half a cubic foot of oxygen combined form one cubic foot of steam. One cubic foot of steam, therefore, which results from the union of these gases, must weigh ·05022 pounds.

A Gang of Steam-Boilers.

STEAM-BOILERS.

Since the introduction of steam as a motive power, a great variety of boilers have been designed, tried, and abandoned; while many others, having little or no merit as steam generators, have their advocates, and are still continued in use. Under such circumstances, it is not surprising that quite a variety of opinions are held on the subject. This difference of opinion relates not only to the form of boilers best adapted to supply the greatest quantity of steam with the least expenditure of fuel, but also to the dimensions or capacity suitable for an engine of a given number of horse-power; the mere arithmetic of the question remaining up to this day unsettled.

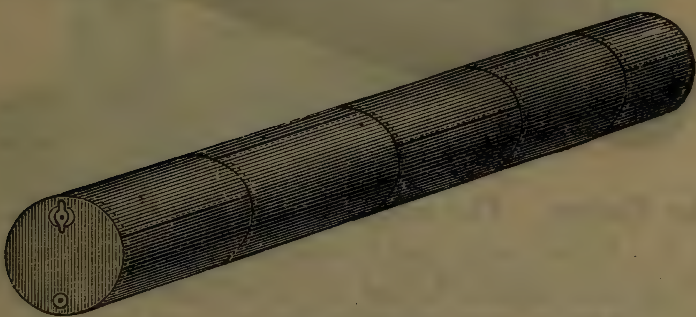
In designing a steam-boiler, there are many important points to be considered, such as cost, proper materials, strength to bear the intended pressure, quantity of steam to be furnished in a given time, space occupied, weight, circulation of water, water room, facilities for cleaning and repairing, steam room, heating and grate surface, area through flues, etc.

The three most important objects to be attained in the design, construction, and use of steam-boilers are, "safety," "durability," and "economy." To insure "safety," it is necessary that the boiler should be designed in accordance with true mechanical principles, avoiding as much as possible the evils of unnatural strains and unequal expansion and contraction; due regard must also be paid to the quality of the material and character of the workmanship employed in its construction.

We have no data by which to establish the general "durability" of any class of steam-boilers, but experience has shown, in all individual cases, that the durability of a steam-boiler depends on the quality of the material and the character of the workmanship used in its construction, the facilities afforded for cleaning, repairing, and renewal of

any of its parts, and also the care and management after being put in use.

"Economy" in the generation of steam depends to a certain extent on the character or quality of the metal of which the boiler is made; as it is a well known fact, that the thicker the iron, and the poorer its conducting qualities, the greater will be the loss of heat; when by using a superior quality of iron, one whose tensile strength and conducting powers are both very great, we lessen the resistance to the passage of the heat from the furnace to the water and greatly increase the economy of the boiler.

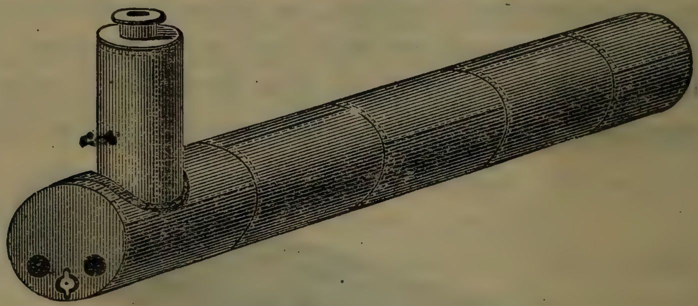


Cylinder Boiler.

It is also well known to engineers that some qualities of iron are two and a half times stronger than others; consequently, if a boiler be made of the poorer iron, that would be as strong as $\frac{1}{4}$ inch of the best iron, it would be necessary to use plates $\frac{5}{8}$ of an inch thick. Even then the heavy boiler would be weaker than the light one, from the fact that the heavy plates would sustain greater injury in the making. In point of economy and durability, the light boiler would be far superior to the heavy one.

Cylinder Boilers.—The plain cylinder boiler, one of the earliest forms of steam-generators, and, until quite recently, the one most extensively used, is fast passing out of use, particularly in localities where space is limited and fuel expensive. Its advantages were its lightness and moder-

ate first cost, and that it afforded better facilities for cleaning, repairing, or the renewal of any of its parts than any other type of boiler. It also possessed peculiar advantages for rolling-mill and blast-furnace purposes, as it required less care, and was least dangerous on account of the great body of water it contained. Its disadvantages were its extreme length and wastefulness of fuel.

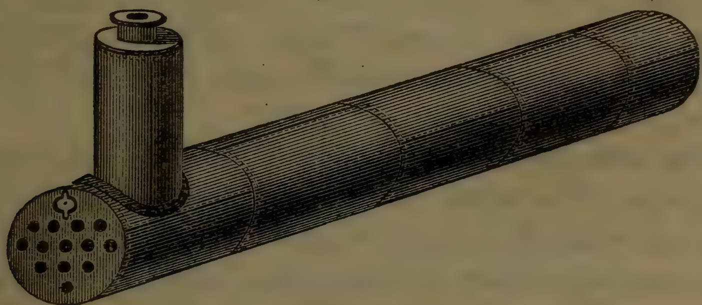


Flue Boiler.

Flue Boilers.—The advantages of this type of boilers over the former are, that it occupies less space, requires less fuel, and steams better in consequence of its extra heating surface. Like the cylinder, it is a favorite for rolling-mill and blast-furnace purposes, as it affords facilities for return draught; but it has the disadvantages of extra weight, consequently, increased first cost, and that it is more difficult to clean or repair. It also requires more care on account of the liability of the flues to become overheated and collapse in case the regular supply of water should be neglected.

Tubular Boiler.—This type of boiler possesses many advantages, in an economical point of view, over either the cylinder or flue, as it occupies less space, requires less fuel to evaporate a certain quantity of water in a given time, and, in consequence of the small diameter of the tubes, its liability to collapse is entirely obviated. But it has the disadvantage of requiring more care than either of the former,

and is almost impossible to clean or repair. The double-deck boiler, a combination of the cylinder and tubular, is a very safe and economical type of boiler, as it occupies less floor space than either the cylinder, flue, or single tubular. It also presents an immense amount of heating surface, and, in consequence of the great body of water it contains, obviates the danger of the water becoming low, excepting in cases of extreme neglect.



Tubular Boiler.

Locomotive Boilers.—This kind of boiler, though not in very general use for stationary purposes, when well proportioned for its work, is very economical, as it occupies but little space, presents an immense amount of heating surface, steams very rapidly, and, when well constructed, is compact and powerful. Its great disadvantages arise from the complication of its parts, which makes it extremely difficult to clean or repair, consequently, it is liable to burn out; besides, as the water space is limited, it requires special care and attention.

STEAM-DOMES.

The advantages claimed to be derived from the steam-dome are, that it acts as a steam reservoir, and also an anti-primer, in consequence of being further removed from the water than any other part of the boiler, which is true

to a certain extent; but as regards its advantages as a steam reservoir, it can easily be shown that an ordinary sized steam-dome adds very little to the steam room of a boiler.

For instance, a boiler 48 inches in diameter and 20 feet long would contain 251 cubic feet of space; if we take $\frac{3}{4}$ of that as water space, we will have left about 63 cubic feet for steam room. Now suppose we take a steam-dome 24 inches in diameter and 2 feet high, we gain only 6 cubic feet of steam room, or about enough of steam to fill the cylinder of an engine 12 inches in diameter and 24 inch stroke, and about 5 times, even if worked expansively.

Now, with respect to its advantages as an anti-primer, it appears to be taken for granted that the higher the point at which the steam is taken from the boiler, the drier it is likely to be; but the cooling effect on the steam, by domes of large diameter exposed to the atmosphere, seems to be entirely lost sight of, as it is a well-known fact that, when an engine is at work, the steam rushes into and through the dome with great velocity, and in its passage is liable not only to take with it a great quantity of water, but have its temperature lowered by coming in contact with so much surface exposed to the action of the atmosphere. It frequently happens that the steam taken from a dome is more wet than that in any other part of the boiler.

The reservoir of power in a boiler is not so much in the steam as in the heated water. With a working pressure of 60 pounds, each cubic foot of steam in the boiler will produce only 4·65 cubic feet of steam at atmospheric pressure; but 1 cubic foot of water in the boiler will produce nearly 35 times that amount, for at 60 pounds pressure the temperature of the water is 307·5°, or 95·5° above the boiling-point at atmospheric pressure; and, as every

degree of heat added to water already at 212° may be taken as competent to generate 1·7 cubic feet of steam, 95·5° will produce 162·35° cubic feet, or nearly 35 times as much as 1 cubic foot of steam at 60 pounds pressure.

It will be seen from the above, that, notwithstanding the general opinion that the presence of a steam-dome is essential for obtaining dry steam and as a remedy for priming, it should be regarded as not only a useless and expensive appendage to a boiler, but a source of real weakness and danger; the practice of cutting a dome-hole in the shell of a boiler, without providing for the weakening of the plate by some other means, should be looked upon as a very mischievous and dangerous practice.

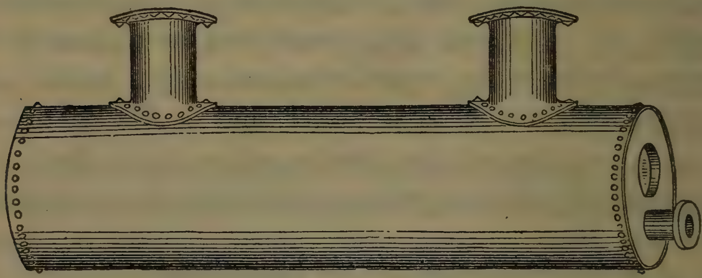
When it becomes necessary to have a dome, as in case of limited steam room, or where the arrangement of the tubes or flues is such as to make it necessary to carry the water high in the boiler, the hole in the plate under the dome should not be cut larger than sufficient to allow a free escape of the steam from the boiler to the dome, or to admit of a convenient adjustment of the dome-braces.

MUD-DRUMS.

When we consider the short life of the mud-drum, which rarely exceeds six or seven years, and also the expense of removing it and replacing it with a new one, its use in any case becomes a question of doubtful economy. Steam users and engineers for a long time entertained the belief that mud-drums were beneficial, inasmuch as they imparted extra heat to the feed-water, and retained the mud that would otherwise have been carried into the boiler. Experience, however, has shown this to be a grave error, as mud-drums impart very little heat to the feed-water, and retain nothing but the earthy matter which is held in suspension in the water, while all the destructive

carbonates that are held in solution are carried into the boiler.

A good deal has been said and written, and many theories advanced, to account for the pitting, or honey-combing, of mud-drums, but the mysterious manner in



Mud-Drums.

which it occurs, and its peculiar character, have not as yet been fully explained, as scientific men are still unable to assign even a plausible cause. But the most probable cause for this singular pitting or rotting away might be assigned to the location of the drum, as it receives nearly all the heat imparted to it on the upper side, with not enough on the lower side to keep the iron perfectly dry and prevent the rusting away of the plates and rivet-heads.

SETTING BOILERS.

In "setting" or "putting in" boilers, as it is sometimes called, all the surface possible should be exposed to the action of the heat of the fire,—not only that the heat may be thus more completely absorbed, but that a more equal expansion and contraction of the structure may be obtained. And in cases where convenience serves, it will be found advantageous to return the draught through a brick flue over the top of the boiler, in order to equalize the heat and, consequently, the expansion; and although this arrange-

ment does not facilitate the generation of steam, yet fuel will be saved by a more complete application of the heat, and the prevention of radiation from the upper part of the boiler.

Convenient openings should be arranged in the brick-work to facilitate the cleaning of the boiler on the outside, as that part of the shell exposed to the action of the draught is liable to become permanently coated with soot and ashes, rendering a great portion of the heating surface nearly worthless; the difficulty experienced in removing these non-conductors from the under side of the boiler generally arises from an improper arrangement at the time of setting, and a want of space.

Boilers should be set with as little brickwork in contact with the shell as practicable. No mortar should be used where it can come in contact with the plates, but fire-clay should be used instead for the whole setting of the boiler.

Long Boilers are often hung by means of loops riveted to the top of the boiler, and connected to cross-beams and arches, resting on masonry above the boiler, by means of hangers. This is a very mischievous arrangement, unless turn-buckles, or some other contrivance, are used to maintain a regular strain on all the hangers, as long boilers exposed to an excessive heat are apt to lengthen on the lower side and relieve the end hangers of any weight; consequently, the whole strain is transmitted to the central hanger, which has a tendency to draw the boiler out of shape, which, in many instances, induces excessive leakage, rupture, and eventually explosion.

The most permanent practical method of setting boilers is to rivet cast-iron brackets or knees to their ends and centre, about 12 feet apart, resting on brick piers, as by that arrangement one end can settle without injuriously

affecting the other. These brackets, in some instances, rest on small rolls arranged in a flanged seat, in order to prevent the piers from being forced out when the boiler expands.

All boilers should be set with an incline of not less than 1 inch in 20 feet to the end in which the blow-off is situated, in order that the water may run out by its own gravity. The blow-off should be located in the back or coldest end, as the mud and deposits always seek that part where the boiling currents are least violent.

EXPANSION AND CONTRACTION OF BOILERS.

A great difficulty to be contended with in the management and working of steam-boilers arises from the unequal expansion and contraction of parts of the structure. In some instances these are so great as to be the cause of more "wear and tear" than any other condition to which the boiler is subjected; consequently, in raising steam on new boilers, or those that have been blown out and allowed to cool down, great care should be taken in allowing the fire to burn moderately, as otherwise the boiler may be seriously weakened, if not permanently injured; more particularly so in the case of flue and tubular boilers, as the flues or tubes, being exposed to the direct action of the fire, and generally of a thinner material than the shell, expand much quicker, and, as a result, the ends of the boiler are forced outward, and the whole structure exposed to an enormous strain.

When the flues of boilers are placed nearer to the bottom than to the top, the strain, from unequal expansion and contraction, is often such that the plates of the under part of the outer shell are torn or broken; and, in other cases, leakages take place in positions where they are most difficult to discover.

When a boiler is set with only a small portion of its bottom exposed to the heat, and a great portion of the structure exposed to the atmosphere, as is the common practice, a powerful action is left at full liberty to work out most injurious results. The heat will expand that portion of the boiler to which it is applied; while the other portion exposed to the cold atmosphere will contract. Thus the two forces are left to exert their respective powers against each other, tending to tear the boiler asunder by means almost imperceptible.

A boiler under steam is often strained, especially in a longitudinal direction, more by the greater dilation of the tubes compared with the shell, or by the unequal expansion of the top and bottom of the shell, than by the actual steam pressure. The persistent leakage often experienced at the seams, along the bottom of horizontal internally fired boilers, might in most cases be ascribed to the difference in temperature of the water and steam at the bottom and top of the boiler; but in some cases the leakage is principally caused by the longitudinal straining of the bottom of the shell, due to the greater expansion of the tubes, especially when the firing is forced in getting up steam after the boiler has been at rest.

As this straining would not take place in testing the boiler by hydraulic pressure in the usual manner, this leakage would not be produced. It follows, from the above considerations, that a hydraulic test might fail to indicate weakness which would be produced and made apparent by steam pressure.

TESTING BOILERS.

Experience has shown that, let a boiler be ever so carefully designed and constructed, there will still remain an element of doubt as to its actual strength, since the

material may have sustained injuries in the process of construction which may have escaped detection.

In the case of a new boiler, even by a first-rate manufacturer, to say nothing of original and hidden flaws in the plates and castings, there is always a possibility of defects, such as bad welding, careless riveting, plates burnt in flanging or cracked in bending, and many other defects that may be traced to reckless negligence or want of skill; consequently, the only means we have of ascertaining, with any degree of certainty, the safety of a boiler is by the application of cold water pressure, as many cases of dangerous defects, which the strictest scrutiny of the practical boiler-maker failed to detect, have been brought to light by means of the hydraulic test.

There are many forms of boilers which do not admit of anything like a proper examination, as, for instance, tubular boilers, in which the shell of the boiler is filled with tubes nearly to the water line; also many forms of marine boilers, whose construction is so irregular and complicated as to defy even an approximate calculation of their strength or condition.

With regard to the various modes of testing by hydraulic pressure, that commonly adopted is to pump water in until the desired pressure be reached. The condition of the joints and rivets is then looked to, and any very conspicuous distortion, leak, or defect marked; and in cases of permanent distortion or flattening of tubes or flues, the injured parts should be immediately removed and repaired or renewed, as the injury to the tube or flue is liable to be aggravated by subsequent tests, and eventually result in rupture or collapse.

Some advocate the method of marking the leaky joints while the pressure is on, and then lowering the pressure for the purpose of calking; this is decidedly wrong. Boilers should never be calked while under steam

or water pressure, however light, as the jarring induced by the calking is liable to spring the seams, and cause fresh leakage in different parts of the boiler.

Some recommend the employment of hot water for testing boilers, as it assimilates, more than cold water, the conditions under which the boiler is placed when at work. But the water for test should never be more than moderately warm, as the hydraulic test is comparatively worthless without a thorough examination of the boiler at the same time; and it is impossible to do so in cases where hot water is used, in consequence of the presence of so much heat under and around the boiler.

In some cases, the plan has been adopted of filling the boiler with water, closing every outlet, and putting fire to it. As water expands about $\frac{1}{24}$ in volume in rising from 60° to 212°, the rise of temperature as the water becomes heated will cause a corresponding increase of pressure, and, from the regularity with which the pressure rises, any leak that may occur in the boiler will be easily noticed by the jerks or starts of the steam-gauge hand. But the wisdom of this method is extremely doubtful, as it involves a certain amount of danger, and prevents the possibility of examining the boiler in the parts most likely to be affected during the test.

In whatever manner a boiler is tested, great care should be taken to obtain the exact amount of pressure employed, for the reason that safety-valves are very often unreliable, particularly so when water pressure is used, and spring-gauges are not always to be trusted under such circumstances; in all cases, when the cold water test is applied, two gauges should be used; for, although a boiler cannot explode under the hydraulic test, yet many serious accidents have occurred by boilers giving way under such circumstances.

The hydraulic test meets with opposition from some engineers on the ground that it does not tell the actual strength of the boiler; but the same objection might be urged against the steam and expansion tests, as there is no accurate method of ascertaining the strength of a boiler but to burst it. The hydraulic test is not meant for perfectly sound boilers, but for the detection of weaknesses in certain parts, and is generally successful for that purpose, if well conducted.

The injuries arising from an excessive application of the hydraulic test are most likely to occur to flue boilers, as, whenever flues subjected to external pressure depart from the true cylindrical form, or form of greatest resistance, they are liable to collapse, even under very low pressure.

Sound Test.—The sound test is generally applied to all accessible parts of the boiler, such as the upper part of the shell, crown-sheet, crown-bars, angle iron, and braces, which is done by tapping the parts to be tested lightly with a small steel hammer. The experienced boiler-maker or inspector can easily tell, by the effect of the sound on the ear, whether the part subjected to the blow is sound or not. But whether by sound, expansion, or hydraulic pressure, the testing of boilers requires the utmost care and experience, and should never be applied to any boiler unless all the conditions are fully understood, such as the diameter, age and condition, character of seams, etc.; nor even then, except by persons who fully comprehend the object and effect of the test.

NEGLECT OF STEAM-BOILERS.

Perhaps no appliances connected with factories, and other places where power is used, are more sadly neglected than steam-boilers, and nothing can be more surprising than this fact, when we consider the important part they

occupy in the manufacturing arts. It would be difficult to assign any reasonable cause for this neglect, except that it may arise from the fact that nearly the whole attention of builders and leading engineers has been concentrated on the improvement and perfection of the steam-engine; and the practical engineer, following the example set by the leaders, generally devotes all his attention to the engine.

In the majority of cases boilers are not cleaned half as often as they should be. When the water is hard, and scale accumulates on the sides or flues of the boiler, solvents are very often resorted to to remove the scale. After the scale has been thrown down, it accumulates on the bottom of the boiler, and, if not removed at once, it becomes conglomerated, forms a heavy coating; and if the boiler is externally fired, the bottom is liable to be burned through.

The yearly report of the Hartford Steam-Boiler Inspection and Insurance Company shows that nearly half of the whole number of defective boilers became so on account of incrustation and deposit of sediment; and, strange as it may seem, there were 40 per cent. more dangerous cases from the deposit of sediment than from incrustation and scale. The same report further shows that more than one-half the defective boilers from other causes are due to careless and incompetent management, proving clearly that a large number of cases from which explosions might be expected may be traced to a direct cause.

CARE AND MANAGEMENT OF STEAM-BOILERS.

Familiarity with steam machinery, more especially with boilers, is apt to beget a confidence in the ignorant which is not founded on a knowledge of the dangers by which they are continually surrounded, but is the offspring of conceit and folly; while contact with steam, and a thorough elementary knowledge of its constituents, theory, and action,

only incline the intelligent engineer and fireman to be more cautious and energetic in the discharge of their duty.

As the boiler is the source of power, and the place where the power to be applied is first generated, and also the source from which the most dangerous consequences may arise from neglect or ignorance, it should attract the special attention of the engineer, as, from the hour it is set to work, it is acted upon by destroying forces, more or less uncontrollable in their work of destruction.

These forces may be distinguished as chemical and mechanical. In most cases they operate independently, yet they are frequently found acting conjointly in bringing about the destruction of the boiler, which will be more or less rapid according to circumstances, care, management, etc.

One of the most common causes of deterioration in steam-boilers, and also leakage of the seams and under side and at the junctions of the tubes and tube sheets, is the reckless practice of blowing out the boiler while still hot, and filling it again with cold water. Under such circumstances, the contraction of the crown-sheet, tube-sheets, and tubes is so rapid and unequal, that, if persisted in, it eventually results in the ruin of the boiler.

Boilers should never be filled with cold water while they are hot, as it has a very injurious effect, causing severe contraction of the seams and stays, which very often induces fracture of stays or leakage in the seams and tubes.

Many boilers, well constructed and of good material, have been ruined by being blown out under a high pressure of steam, and then suddenly filled with cold water.

The tubes of boilers being generally of thinner material than the shell, consequently cool and contract sooner; for this reason, the boiler should never be filled with cold water while the tubes are hot.

The boiler should always be allowed to stand for several

hours, or until it is cold, before the water is run out; the deposit of mud and scale will then be found to be quite soft, and can be easily washed out with a hose from all accessible parts.

There seems to be an impression on the minds of some engineers that blowing out a boiler under pressure has a tendency to remove the deposits of mud from the boiler; but experience has shown this to be a very grave mistake.

Tubes and flues should be frequently swept out, at least once a week. This can be done, in either land or marine boilers, while the engine is in motion, by covering the fire in the furnace, either in front of or under the tubes to be cleaned, with a thick layer of fresh coal, and cleaning one set of tubes at a time. Accumulations of salt frequently occur in the tubes of marine boilers, which are induced by leakage of the tubes or tube-sheet; such accumulations should be removed as soon as discovered, and the tube thoroughly swept, or, if need be, bored out with a steel scraper, in order to prevent the part from becoming burnt through. It frequently becomes necessary to direct a steam-jet on the deposit before it can be effectually removed.

Tubes frequently become leaky in consequence of becoming split, or their ends at the tube-sheet being burned off. In the former case they have, of necessity, to be plugged, which can be done effectually by driving pine plugs solid into their ends; although, in some cases, it becomes necessary to run a long bolt through the inside of the tube, with a flange and packing at each end; in the latter case, they can be made tight by means of wrought-iron ferrules.

When two or more boilers are connected by feed-pipes, the stop-valves on each should be shut off every night, or whenever they are not working, as the water is

liable to escape from one to the other, on account of variation in the pressures; and, as a consequence, when the water in one is up to, or even above, the proper level, the tubes or flues in the other are very often bare of water.

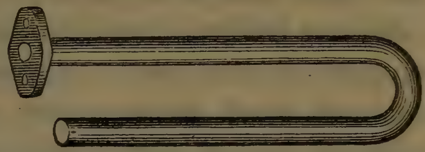
It is no uncommon thing in factories to have two boilers for the same engine, in order that one may be out of use while the other is working; but, while this is an accommodation, it is not always economy, as boilers wear out faster when not in use, by oxidizing and corroding, than if moderately worked. It will be found more economical to work with extra boiler room than to have one or more standing. It will also tend to prevent priming. The furnaces will be more economically worked with a thick fire than with a thin one, by allowing the heat to accumulate, thereby maintaining a high temperature in the furnace with slow combustion.

The furnace door should never be allowed to remain open longer than a sufficient time to clean and replenish the fire, as the contraction of the tubes and flues, induced by the cooling down of the furnace, has a very mischievous effect on all parts of the boiler exposed to the cold draught, more particularly so when the fire is thin, and the temperature in the furnace is of a high degree.

The feed-water should be sent into the boiler as hot as possible, as, if it be forced in at a low temperature, it will impinge on that portion of the boiler with which it comes in contact, and, as a result of the continual expansion and contraction induced by the varying temperature of the water, the boiler is liable to crack and become leaky.

Where economy of fuel is no object, as is often the case at coal-mines, saw-mills, and wood-working establishments, a very inexpensive way of averting the disastrous effects of pumping cold water into boilers is to introduce

the feed-pipe into the back end of the boiler, carrying it forward about three-quarters the length of the boiler, and then returning it to the back end where the water is discharged into the boiler. By this arrangement the water will have a temperature nearly equal to that of the water in the boiler when discharged from the pipe.



Heater Pipe.

If, from neglect or any other cause, the water in the boiler should become dangerously low, the fire-doors and damper should be immediately thrown open for the purpose of admitting the cold air to the heated plates, and the fire withdrawn as soon as possible. Under such circumstances, no attempt should be made to introduce cold water into the boiler, as it might be attended with the most disastrous results. Nor should the safety-valve be tampered with, as any moving of the safety-valve under such circumstances would have a tendency to cause violent agitation or foaming of the water over the hot plates, which would have the effect of generating more steam than the safety-valve could discharge, and most likely result in an explosion.

Every boiler should be furnished with a safety-valve of sufficient capacity to prevent the steam from getting to a higher pressure than that considered safe, and the length of the lever should be as short as possible, in order that the pressure may not be increased with the same weight.

The safety-valve should always be moved before the fire is started to get up steam, in order to ascertain if it is in good working order. It should also be raised whenever the boiler is being filled with cold water, so as to allow the air to escape, as air has a tendency to retard the influx of the water, and also to occupy the steam room when

steam is raised. Air also interferes with the uniform expansion of the boiler. The safety-valve should be kept open until the steam commences to escape.

All new boilers should be thoroughly examined before being filled with water, in order to ascertain if there are any tools, wood, lamps, greasy waste, etc., left behind by the boiler-makers, that would be liable to be carried into the connections, or cause the boiler to foam.

In getting up steam in boilers just filled with cold water, or that have been out of use for sometime, the fire should be allowed to burn moderately at first, in order to admit of the slow and uniform expansion of all parts of the boiler; as, when the fire is allowed to burn rapidly from the first start, some parts become expanded to their utmost limits, while others are as yet nearly cold, thereby exposing the boiler to those fearful strains induced by unequal expansion and contraction, resulting, as they always do, in leakage, fracture, and sagging of the shell and flues.

In all cases, the engineer, or the person having charge, should ascertain with certainty the height of the water in the boiler before opening the draught or starting the fire, as any neglect to do so might be productive of great danger and inconvenience.

When boilers are laid up, or out of use, even if it be for a few days, they should be opened, cleaned, and thoroughly examined, in order to ascertain if any of the stays or braces have become loose or slack, or disconnected. Before being closed up, all gaskets for man- and hand-holes, and grummets for mud-holes, should be painted with a coating of black lead, in order to protect their seats from deterioration, induced by the chemical action of the sulphur in the gum packing, now so universally used for the joints of steam-boilers.

In view of the above enumerated evils that so silently

and persistently affect the durability of steam-boilers, the question might naturally be asked, "What guarantee of safety have steam users and the public against disastrous explosions?" The answer would be, that safety does not depend so much on the strength of boilers as it does on their care and management, from the fact that a thorough knowledge of their condition enables intelligent engineers to avoid numerous causes and remedy many defects that would ultimately lead to destruction.

HEATING SURFACE.

The evaporative power of a boiler mainly depends upon the efficiency of its heating surface, whose duty it is to transfer the heat from the products of combustion without to the water within.

The heat is communicated to the transmitting surface in two different ways,—by radiation and by contact; and from two or three different hot masses in the furnace, viz., the solid incandescent fuel, the flame, and the hot gases produced by combustion. Beyond the furnace-bridge or tube-plate, the heat is imparted by contact and radiation from the flame and gases only.

The amount of heat transmitted by radiation from one body to another diminishes as the square of the distance between the bodies increases. The effect on any surface is also diminished by any increase in the inclination at which the rays fall upon it.

The radiation from solid incandescent fuel is greater than from flame, whilst transparent hot gases scarcely radiate any heat at all. The more intense the contact heat of the flame by thorough mixture with the air, the less is the heat by radiation.

Conduction is the transfer of heat either between the particles of the same body, or between the parts of dif-

ferent bodies in contact, and it is distinguished respectively as internal and external conduction. The rate at which the former takes place in metal plates is very much greater than the latter, where the heat passes from the hot mass to the plates, and from these again to the water.

The efficiency of any heating surface may be defined as the proportion borne by the amount of heat it transmits to the whole amount available for transmission.

A flat, horizontal surface, not too far above the layer of fuel, is usually considered to be the most favorable for raising steam. By being made concave to the fire, it has, however, the further advantages of being still better adapted for receiving the radiant heat; of facilitating the access of fresh supplies of water to replace the heated ascending particles, and thereby promoting the circulation; of boiling off the matters deposited from the water, and so preventing incrustation; and of being stronger, and in some cases more durable.

Next in efficiency to the flat surface with the water above, comes the sloping surface surrounding the fire, which is superior to one in a vertical position, as it receives the rays of heat at a more favorable angle, and allows the steam bubbles to escape more freely. The value of horizontal surfaces beneath the fire is not worthy of consideration as heating surface.

In externally fired boilers the heating surface is usually convex to the fire. This is, by many, regarded as inferior to a concave surface, probably because it is not so well adapted for directly receiving the radiant heat from the fire, and does not appear to offer an equal facility for circulation. The results obtained from this description of surface in actual work do not appear to verify this conclusion. The inferior evaporative power usually alleged of the ordinary externally fired boiler is, in a great measure, due to the waste of heat in the furnace.

RULES FOR FINDING THE HEATING SURFACE OF STEAM-BOILERS.

Rule for Locomotive or Fire-box Boilers.—Multiply the length of the furnace-plates in inches by their height above the grate in inches; multiply the width of the ends in inches by their height in inches; also, the length of the crown-sheet in inches by its width in inches; multiply the combined circumference of all the tubes in inches by their length in inches; from the sum of the four products subtract the combined area of all the tubes and the fire-door; divide the remainder by 144, and the quotient will be the number of square feet of heating surface.

Rule for Flue Boilers. — Multiply $\frac{2}{3}$ of the circumference of the shell in inches by its length in inches; multiply the combined circumference of all the flues in inches by their length in inches; divide the sum of these two products by 144, and the quotient will be the number of square feet of heating surface.

Rule for Cylinder Boilers. —Multiply $\frac{2}{3}$ of the circumference in inches by its length in inches; add to this product the area of one end; divide this sum by 144, and the quotient will be the number of square feet of heating surface.

Rule for Tubular Boilers. —Multiply $\frac{2}{3}$ of the circumference of the shell in inches by its length in inches; multiply the combined circumference of all the tubes by their length in inches. To the sum of these two products add $\frac{2}{3}$ the area of both tube-sheets; from this sum subtract twice the combined area of all the tubes; divide the remainder by 144, and the quotient will be the number of square feet of heating surface.

EVAPORATIVE EFFICIENCY OF BOILERS.

The evaporative efficiency of a given amount of heating surface depends upon the time allowed for the transmission of heat through it, or for the contact of the hot gases. The greater their velocity, the less time they have to impart their heat to the plates or tubes where the length of the surface is constant. The velocity through a tube may be increased either by reducing its area, the total quantity of gases passing through remaining constant, or by increasing their draught, and so causing a greater amount of gases to pass through in a given time, the area of the tube remaining unaltered.

When the heating surface consists chiefly of tubes, as in the locomotive type of boiler, the collective area of the tubes may be diminished without decreasing the extent of heating surface, since the sectional area varies as the square of the diameter, whilst the surface measured by the circumference diminishes simply as the diameter. With the gases passing at the same velocity through two tubes whose diameters are as 1 : 2, the latter will be traversed in a given time by four times the quantity of gases, and will have only twice the surface to absorb the heat.

Therefore, to obtain the same evaporative economy as in the small tube, we must double the length of the larger, or, generally speaking, the proportion between diameter and length of a tube is constant for the same evaporative efficiency.

When an increased quantity of gases of the same density pass through a tube in a given time, although there will be a greater absorption of heat, there will still be a loss by the increased amount of heat remaining in the escaping gases; and in order to preserve the same economy, or in order that the heat of the escaping gases shall remain constant, the length of the tube must be increased in

proportion to the increased quantity of gases passed through.

When we consider the heat to be imparted to the tube surface by radiation, which, however slight, is probably the principal mode of transfer in vertical and other long tubes, where the convection among the particles of gas cannot be supposed to take place to any great extent, we may assume the heat to be concentrated in the axis of the tube, whence we find the quantity of heat received in a given time by the surface from radiation will be inversely as the square of the diameter.

By doubling the diameter we shall have four times the quantity of gases passed through, and the quantity of heat received in a given time will be only one-quarter of what it was before, owing to the increase of distance.

The surface being, however, twice as great, the absorption per unit of length becomes equal to the original. Therefore, in order to bring the evaporative efficiency up to the original, we must double the length of tube, or generally we must increase the heating surface as the square of the diameter, in order to obtain the same evaporative efficiency from radiation when increasing the diameter of a tube.

But if we reduce the diameter to one-half, we increase the absorbing power fourfold per unit of surface; the heating surface being, however, reduced to one-half, the evaporative power of the tube will be only doubled, whence the tube may be reduced to one-half the original length and still retain the same evaporative efficiency, or, the length remaining unaltered, the quantity of gases passing through should be doubled to maintain the same temperature at the escaping end, or, as before, the efficiency of each square foot of heating surface increases inversely as the square of the diameter.

The evaporative efficiency of a square foot of heating surface varies in different classes of boilers, as well as in the same boiler under different conditions; in consequence of this there is considerable difficulty in determining the precise area of heating surface necessary for the production of a given amount of steam in a given time.

For a given description of boiler, it is evident the evaporative efficiency will mainly depend upon the ratio between the quantity of coal consumed and the extent of heating surface, as well as the quality of the fuel and the manner in which it is burned.

The easiest method, and consequently the one most frequently adopted, is to measure the quantity of water by the difference of its height in the glass gauge at the beginning and end of the experiment. But this method is very uncertain, as there can be but little doubt that in many boilers the surface of the water is not level, but is generally higher over the furnace, where the greatest ebullition takes places; the difference in the height at any time will greatly depend on the intensity of the firing.

Meters are frequently employed for measuring the quantity of water that enters a boiler in a given time; but, like all other contrivances resorted to for that purpose, they are not always reliable. The only sure method of ascertaining the quantity of water evaporated is by actual measurement with a cistern or vessel whose cubic contents are accurately known. The quantity of water in the boiler before and after the trial should be measured at the same temperature, which should not exceed 212°, to insure accuracy.

But even when the amount of water introduced, and the quantity passed off from the boiler, are accurately ascertained, there yet remains a doubt as to how much has been actually evaporated, and how much may have passed off

in priming, as there are very few boilers that do not prime more or less, and the quantity of water passed off in this manner is sometimes very considerable, and often furnishes boiler-makers, and more particularly manufacturers of patent boilers, an opportunity to delude steam users with the belief that their boilers are capable of evaporating 14 or 15 pounds of water to 1 pound of ordinary coal.

In fact, unless the amount of water passed over with the steam by priming, when working under pressure, can be accurately ascertained, it is utterly impossible to determine the evaporative capacity of the boiler.

HORSE-POWER OF BOILERS.

It must be admitted that the manner in which the power of a boiler is usually calculated is far from satisfactory. As it has long been the custom to estimate boilers by their real or nominal horse-power, and the nominal horse-power of engines is usually based upon the diameter of the cylinder, without regard to other conditions, so in boilers the nominal standard of power is estimated by their size, without regarding the pressure of steam, the efficiency of heating surface, size of grate, rate of combustion, quality of fuel, etc.

At the present time, it might be said that there is no received rule for estimating the power of a steam-boiler; that is to say, no rule generally recognized by the trade. As it has long been a custom in England to estimate the horse-power of boilers for stationary engines by their length, regardless of the diameter and other conditions, if we turn to marine engines, we find some makers estimating their power entirely by the grate surface; but while one maker divides his grate area by ·8 and calls the result the horse-power, another uses ·75, and another uses ·5. Thus, a boiler with 100 square feet of grate sur-

face may be called 125-, 133-, or 200-horse power. Others, again, neglect grate surface altogether and go by heating surface, and anything between 12 and 25 feet is said by different makers to represent a horse-power.

In view of the foregoing facts, it is very desirable that, in purchasing boilers, some understanding should be established as to the quantity of steam they are capable of furnishing in a given time.

An idea has been very generally entertained by boiler-makers and boiler dealers that it would be impossible to lay down any rule that would apply to all classes and varieties of boilers; but this seems quite improbable, as there should be no great difficulty in dividing boilers into different classes, and establishing, as a rule, the number of square feet of heating surface, steam room, water space, and grate surface that would form a standard of horse-power for each class.

And although it might be somewhat difficult to establish a standard that would apply with strict accuracy to all classes of boilers, still it might be made approximately so.

Certain vague notions have long existed among engineers and steam users that, in adjusting the dimensions of steam-boilers, it is better to have them larger than is absolutely necessary, consequently, it has grown into a custom to recommend a 15-horse power boiler for a 10-horse power engine, a 25-horse power boiler for a 20-horse power engine, and so on.

This rule works well for the seller, but it does not always work both ways, as boiler-makers very often deceive purchasers in the extent of heating surface that they ought to receive, such misrepresentations giving rise to general distrust, disappointment, and dissatisfaction between manufacturers and purchasers. It seems rather

singular that all the ordinary methods of trade should be changed when the question becomes one of the purchase or sale of a steam-boiler.

The rule most common in use in this country, if it might be called a rule, for determining the horse-power of steam-boilers, is to estimate the entire heating surface, and give an allowance for a horse-power; for cylinder boilers, 14 square feet of heating surface, and 1 square foot of grate surface; for flue boilers, 15 square feet of heating surface, and $\frac{3}{4}$ of a square foot of grate surface; and for tubular boilers, 15 to 16 square feet of heating surface, and $\frac{1}{2}$ of a square foot of grate surface.

A more liberal allowance of grate surface for the flue and tubular boilers would give more satisfactory results, as, when the grate surface is limited, the fuel has necessarily to be exposed to a sharp draught, which induces a great loss of the heated gases, as they are carried into the chimney without having sufficient time to impart their heat to the flues and tubes.

EXAMPLE.

Diameter of boiler, 48 inches.
Length " " 11 feet.
46 3-inch tubes.

```
3)150·7968 circumference of shell.
  --------
   50·2656
         2
  --------
  100·5312
       132 length in inches.
  --------
   2010624
  3015936
 1005312
 ----------
 13270·1184 sq. in. heating surface in shell.
```

```
      9·4248 circumference of 1 tube.
          46
    --------
      565488
     376992
    --------
    433·5408
         132
    --------
     8670816
   13006224
   4335408
  ----------
  57227·3856 sq. in. heating surface in tubes.
3)1809·5616 area of 1 tube-sheet.
   603·1872
          2
   ---------
  1206·3744
          2
  ----------
  2412·7488 sq. in. heating surface in tube-sheets.
  7·0686 area of 1 tube.
      46
  -------
  424116
 282744
 --------
 325·1556 area of tubes.
        2
 --------
 650·3112
 13270·1184
 57227·3856
  2412·7488
 ----------
 72910·2528
   650·3112
 ----------
144)72259·9416 total heating surface in sq. in.
  16)501·8     sq. ft. of heating surface.
     31·       horse-power.
```

Grate surface required, 15½ square feet.

FIRING.

Firing, like engineering, ought to be recognized as a profession, and none but intelligent men, who can appreciate the importance of their position, should be placed in

charge of the coal pile; as it is a well-known fact that when the engineer has done all he can to attain economy in the steam-engine, much of the result still remains in the hands of the fireman.

The use of a more improved class of steam-engines involves the necessity of employing more skilful and careful attendants; not that the work is more difficult, as less coal has to be thrown into the furnace, but because a careless or unskilful fireman can counteract all the ingenuity displayed in the improvement, construction, and management of the engine.

Consequently, every engineer should be required to prepare himself for the duties of his profession by commencing as a fireman; otherwise, how can he be expected to be able to instruct his fireman in the manner of firing best calculated to insure the most satisfactory and economical results?

Clean grate-bars, with an even distribution of the fuel in the furnace, the exercise of judgment in the quantity of air admitted, and the regulation of the draught, are the main points to be attended to; and although they require the exercise of skill and intelligence, they cannot be said to involve an unreasonable amount of either labor or vigilance.

Even with the best coal and most careful firing, a quantity of the coal falls through the fire-bars either as unburnt coal or ashes. Another portion goes up the chimney, unconsumed, in the form of smoke and soot; and a further quantity, half consumed, in the form of carbonic oxide. The loss from these causes may amount to from 2 to 20 per cent. It all arises from wrongly constructed furnaces and bad firing, and can nearly all be avoided.

Most coal contains a greater or less quantity of moist-

ure, and the evaporation of this moisture causes the first loss of heat. Radiation from the furnace causes a further loss. But the great causes of loss are the admission into the furnace of a large quantity of useless air and inert gases, and the escape of these, with the actual products of combustion, up the chimney, at a very much higher temperature than that at which they entered the furnace.

Air is composed of about one-third oxygen and two-thirds nitrogen. The oxygen only is required to effect the combustion of the fuel, and the useless nitrogen merely abstracts heat from the combustibles, and lowers the temperature of the furnace. About 12 pounds of air contain sufficient oxygen to effect the combustion of 1 pound of coal, but, owing to the difficulty of bringing the carbon into contact with the oxygen, the quantity actually required to pass through the furnace is from 18 to 24 pounds of air per pound of coal burnt. The surplus air passes out unburnt, and its presence in the furnace lowers the temperature subsisting there, and abstracts a portion of the heat generated.

As the whole of the air enters the furnace at about 60° Fah., and the unconsumed air and products of combustion leave the flues at from 400° Fah. to 800° Fah., the total loss from these causes is from 20 to 50 per cent. Each pound of good coal burnt is theoretically capable of evaporating about 15 pounds of water; in good practice it evaporates but 9 or 10 pounds, and in ordinary practice but 6 or 8 pounds of water.

There are difficulties in the way of abstracting all the heat from the furnace gases; first, because, with natural or chimney draught, the gases require to pass into the chimney at not less than 500° Fah., in order to maintain the draught; and secondly, because the transmission of heat from the gases to the water, when the difference of their

temperatures is small, is so slow that an enormous extension of the surface in contact with them becomes necessary in order to effect it.

But by having energetic combustion and a high temperature in the furnace, the quantity of air actually required may be much reduced; by suitable arrangements for admitting air and feeding coal into the furnace, the proportions of each may be suitably adjusted to each other; and by a liberal allowance of properly disposed heating surface, the temperature of the reduced quantity of furnace gases may be reduced to that simply necessary to produce a draught in a furnace with natural draught, or to about 400° Fah., or less, in a furnace where the draught is obtained from a steam jet or fan.

There have not been, heretofore, that attention and thought devoted to the examination of the subject of the economy of fuel which the magnitude of the interest involved and its importance in a national point of view render it worthy of. The saving of one pound of water per horse-power per hour for ten hours a day, providing the engine is 100-horse power, and assuming that the boiler evaporates 7 pounds of water per pound of coal, would make a saving of 1000 pounds of water per day, which would require the consumption of 143 pounds of coal per day, or $22\frac{1}{2}$ tons a year, the cost of which would be, at the ordinary price of coal, over $125.

The methods most in vogue for the consumption of all kinds of fuel are those which gradually developed themselves, as necessity dictated, to the untutored intellect of uncultivated men, and which, however creditable to the men that devised them, inasmuch as they availed themselves of all the sources of information within their reach, are nevertheless a reproach to the more advanced knowledge of physical and mechanical science enjoyed by the present generation.

INSTRUCTIONS FOR FIRING.

In estimating the relative merits of different steam-engines, it is generally assumed that the fuel is burned under conditions with which the men who supply coal to the furnaces have nothing whatever to do,—in short, that any man who can throw coal on a fire and keep his bars clean must be as good as any other man who can do apparently the same thing.

But this conclusion is totally erroneous, as it is within the experience of nearly every engineer and steam user that many engines now in operation throughout the country consume twice as much fuel, per horse-power, as is required in those that are more economically managed.

When a boiler is of sufficient capacity to generate the necessary amount of steam without urging the fires, it will be found most advantageous to carry a thick bed of coal on the grates, as, when the coal can be burned in large quantities and with a moderate draught, the heat is more generally utilized than if the coal is burned in small quantities and with a sharp draught.

For stationary boilers the fuel should not be less than from three to four inches thick on the grate. For marine boilers, if anthracite coal be used, from 5 to 6; if bituminous, from 6 to 8 inches. Of course, the thickness of the fire must be governed by the character of the fuel and quantity of steam required.

Before starting a fresh fire in the furnace, a thin layer of coal should be scattered over the grate; most of the kindling, whether shavings, oily-waste, or paper, should be placed on the ends of the bars next the door, and then covered with a uniform layer of wood. This is a necessary precaution, as, when the fuel fails to ignite at the front at

first, it generally takes a long time before the fire burns through.

When the coal is in large lumps, so that the spaces between them are of considerable size, the depth may be greater than where the coal is small and lies compactly; and where the draught is very strong, so that the air passes with great velocity over and through the fuel, there is not time for the carbonic acid to combine with and carry off the products of combustion, and consequently a bed of greater depth may with propriety be used.

When very large coal is used, it will be found of immense advantage to mix it with some small coal; more particularly so, when the draught is strong, as such an arrangement forms a resisting barrier to the currents of cold air that would otherwise pass through the interstices between the lumps, and render the combustion more perfect.

When an increasing quantity of steam is wanted, the average thickness or quantity of fuel on the grate must not be increased, but rather diminished, and supplied in smaller quantities and more frequently. As soon, however, as the supply of steam exceeds the demand, the coal may again be supplied in larger quantities at a time.

In firing up, the coal should be scattered evenly over the grate, but thinner at the front near the dead-plate than at the middle or back, and no portion of the grate should ever be left uncovered.

When it becomes necessary to replenish the fire, it should be done as quickly as possible, as, when the damper and the fire-door are both open at the same time, the current of cold air passing through the furnace above the fuel not only reduces the temperature in the furnace, but has a tendency to injure the boiler.

There should in all cases be ample fire in the furnace,

an extra quantity of water in the boiler, and a full head of steam, before any attempt is made to clean the fire; then the damper should be opened to its full limit, in order that the heated gases and dust may pass into the flue; and, if there be more than one fire, one only should be cleaned at a time, and allowed to become thoroughly kindled before the next one is cleaned.

The fire should never be allowed to become low for the purpose of making it more easy to clean, as, in consequence of the small quantity of fire in the furnace after cleaning, it would have a tendency to go nearly out, which is often attended with great loss and inconvenience. It is always best to have a good fire, then close the damper and open the furnace door, in order to take the white glare off the fire before commencing to clean it; the damper should then be reopened to its full extent and all the live fire pushed back to the bridge, without disturbing any of the ashes or cinders; the latter should then be drawn out, and the fire that was pushed back, drawn forward to one side, and the ashes and cinders that remain near the bridge removed. The fire should then be distributed evenly over the grate, all the cinders and clinkers that remain picked out, and the fire covered with a thin layer of fresh coal, care being taken to waste none of the combustible fuel.

Before commencing to clean the fire, it is always advisable for the fireman to place a piece of scantling a short distance in front of the furnace, in order to protect his feet from the hot cinders as they fall out.

In cleaning the fires of locomotive, marine, or other fire-box boilers, water should not be thrown in the ash-pit, as the lye formed from the wet ashes has a tendency to corrode and destroy the fire-box and water-legs.

The fire should never be disturbed so long as any

light shines through the grate into the ash-pit, unless the boiler fails to furnish the necessary amount of steam. Even then it is better, if anthracite coal be the fuel, to shed out the ashes from the bottom through the grate with a thin hooked poker; but if bituminous coal be used, it requires frequent breaking up, in order to allow the air to intensify the combustion. When broken up, it should always be pushed back toward the bridge, and the fresh fuel supplied in the front and allowed to coke. The smaller the quantity supplied at a time, and the more attention paid to its distribution and regulation, the more perfect will be the combustion and the more intense the heat.

If, from neglect or any other cause, the fire should become very low or the grate partly stripped, it should not be poked or disturbed, as that would have a tendency to put it entirely out; but wood, shavings, saw-dust, greasy waste, or some other combustible substance, should be thrown on the bare places, and, after being covered with a thin layer of coal, the damper opened to its full extent.

If strict attention be paid to the regulation of the furnace, and coal applied to only one side of the fire at a time, nearly all the smoke can be consumed and quite a saving in fuel effected. Fresh coal should never be supplied except when absolutely necessary, and even then only in small quantities and at such places as are most affected by the draught, as it is a common error, with inexperienced firemen, to continually supply coal to the furnace, which eventually becomes choked, and the combustion of the fuel rendered imperfect.

The regulation of the draught should receive particular attention, as air costs nothing, while fuel is quite expensive; therefore none of the latter should be allowed to pass out of the furnace without being fully utilized. The ash-pit and front of the furnace should at all times be kept free

from dirt, ashes, and cinders, as such accumulations have not only the effect of diminishing the cubic contents of the space under the furnace, but also of obstructing the free flow of air through the grate-bars, so essential to the perfect combustion of the fuel.

It is a well-known fact, that much of the waste attributed to the steam-engine occurs in the furnace, and while some of it may be unavoidable, a great portion of it, nevertheless, is due to bad firing, which is the result of ignorance, carelessnes, or inattention.

RULES FOR FINDING THE QUANTITY OF WATER BOILERS AND OTHER CYLINDRICAL VESSELS ARE CAPABLE OF CONTAINING.

Rule for Cylinder Boilers.—Multiply the area of the head in inches by the length in inches, and divide the product by 1728; the quotient will be the number of cubic feet of water the boiler will contain.

EXAMPLE.

Diameter of head, 36 inches.
Area " " 1017·87 "
Length of boiler, 20 feet, or 240 inches.

```
        1017·87
            240
        -------
        4071480
       203574
       -------
1728)244288·80
       --------
        141·37 cubic feet.
```

Rule for Flue Boilers.—Multiply the area of head in inches by the length of the shell in inches; multiply the combined area of the flues in inches by their length in inches; subtract this product from the first and divide the

remainder by 1728; the quotient will be the number of cubic feet of water the boiler will contain.

Rule.—*To find the Requisite Quantity of Water for a Steam-boiler.*—Add 15 to the pressure of steam per square inch; divide the sum by 18; multiply the quotient by ·24; the product will be the quantity in U. S. gallons per minute for each horse-power.

Rule.—*To find the Required Height of a Column of Water to supply a Steam-boiler against any given Pressure of Steam.*—Multiply the boiler pressure in pounds per square inch by 2·5; the product will be the required height in feet above the surface of the water in the boiler.

Another Rule.—*To find the Requisite Quantity of Water for a Steam-boiler.*—When the number of pounds of coal consumed per hour can be ascertained, divide it by 7·5, and the quotient will be the required quantity of water in cubic feet per hour.

LONGITUDINAL AND CURVILINEAR STRAINS.

The force tending to rupture a cylinder along the curved sides depends upon the diameter of the cylinder and pressure of steam; and we may regard, hence, the total pressure sustained by the sides to be equal to the diameter × pressure per unit of surface × length of cylinder, neglecting any support derivable from the heads, which, in practice, depends on the length.

It must be understood that the strain on a boiler subjected to internal pressure transversely, is exactly double what it is longitudinally, or in other words, the strain on the longitudinal seams is double that on the curvilinear. And no matter what the diameter of a boiler may be, the transverse pressure tending to tear it asunder will always be double the pressure exerted on the curvilinear seams.

RULES.

Rule *for finding Safe Working Pressure of Iron Boilers.*—Multiply the thickness of the iron by ·56 if single riveted, and ·70 if double riveted; multiply this product by 10,000 (safe load); then divide this last product by the external radius (less thickness of iron): the quotient will be the safe working pressure in pounds per square inch.

EXAMPLE.

Diameter of boiler.......................................42 inches.
Thickness of iron.. $\frac{3}{8}$ inch.

```
              2)42
                ——
                21 external radius.
               ·375
             ——————
            20·625 internal radius.
Thickness of iron ⅜ = ·375
                ·56 single riveted.
              ——————
               2250
              1875
              ——————
              ·21000
                10000 safe load.
              ———————————
   20·625)2100·00000
          ———————————
              101·81 pounds safe working press.
```

In the above rule, 50,000 pounds per square inch are taken as the tensile strength of boiler iron, and one-fifth of that, or 10,000, as the safe load. Hence five times the safe working pressure, or 50,000 pounds, would be the bursting pressure.

Rule *for finding the Safe Working Pressure of Steel Boilers.*— Multiply the thickness of steel by ·56 if single riveted, and ·70 if double riveted; multiply this product by 16,000 (safe load); then divide this last product by the external radius (less thickness of steel): the quotient will be the safe working pressure in pounds per square inch.

EXAMPLE.

Diameter of boiler.. 44 inches.
Thickness of steel.. ¼ inch.

$$\begin{array}{r l}
2)\underline{44} & \\
22 & \text{external radius.} \\
\underline{\cdot 25} & \\
21\cdot 75 & \text{internal radius.}
\end{array}$$

Thickness of steel ¼ = ·25

$$\begin{array}{r l}
\underline{\cdot 70} & \text{double riveted.} \\
1750 & \\
\underline{16000} & \\
1050000 & \\
175 & \\
21\cdot 75)\overline{2800\cdot 000} & \\
128\cdot 73 & \text{safe working pressure.}
\end{array}$$

80,000 being taken, in the above rule, as the tensile strength of steel, and one-fifth of that, or 16,000, as the safe load. Hence 80,000 would be the bursting pressure.

Rule *for finding the Aggregate Strain caused by the Pressure of Steam on the Shells of Steam-boilers.*—Multiply the circumference in inches by the length in inches; multiply this product by the pressure in pounds per square inch. The result will be the aggregate pressure on the shell of the boiler.

EXAMPLE.

Diameter of boiler.. 42 inches.
Circumference of boiler.......................... 131·9472 "
Length " .... 10 ft., or 120 "
Pressure " .... 125 pounds.

131·9472 × 120 × 125 = 1,979,208 pounds ÷ 2000 = 989 tons.

EXPLANATION OF TABLES OF BOILER PRESSURES ON FOLLOWING PAGES.

The figures ⅜, 00, 0, 1, etc., in the horizontal column on the top of the tables on pages 390, 391, 392, 393, 394, 395, 396, and 397, represent the number of the iron or steel.

The decimals in the second horizontal column are equal to the fractional parts of inches in the third column. The vertical column on the left-hand side represents the diameter of the boiler in inches. All the other columns represent pounds pressure.

Example.—24-inch diameter, $\frac{3}{8}$ steel, 289·03 pounds per square inch.

TABLE

OF SAFE INTERNAL PRESSURES FOR IRON BOILERS.

Birmingham Wire Gauge.		$\frac{3}{8}$	00	0	1	2
Thickness of Iron.		.375 $\frac{3}{8}$	.358 $\frac{3}{8}$ Scant.	.340 $\frac{11}{32}$	.300 $\frac{5}{16}$	.284 $\frac{9}{32}$
	Dia. In.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.
External Diameter.	24	180.65	172.20	163.29	143.59	135.75
	26	166.34	158.58	150.39	132.28	125.08
	28	154.13	146.96	139.38	122.63	115.95
	30	143.59	136.92	129.88	114.29	108.07
	32	134.40	128.17	121.58	107.01	101.20
	34	126.31	120.47	114.29	100.60	95.14
	36	119.15	113.64	107.81	94.92	89.77
	38	112.75	107.54	102.04	89.84	84.98
	40	107.01	102.07	96.85	85.28	80.67
	42	101.81	97.12	92.11	81.16	76.77
Longitudinal Seams, Single Riveted.	44	97.11	92.63	87.90	77.42	73.24
	46	92.82	88.54	84.02	74.01	70.01
	48	88.89	84.80	80.47	70.89	67.06
	50	85.28	81.36	77.21	68.02	64.35
	52	81.95	78.18	74.20	65.37	61.84
	54	78.87	75.25	71.42	62.92	59.53
	56	76.02	72.53	68.84	60.65	57.38
	58	73.36	70.00	66.43	58.54	55.38
	60	70.89	67.63	64.19	56.57	53.52
	62	68.57	65.43	62.10	54.72	51.78
	64	66.40	63.36	60.14	53.00	50.15
	66	64.37	61.42	58.30	51.38	48.61
	68	62.45	59.59	56.57	49.85	47.17
	70	60.65	57.87	54.93	48.41	45.81
	72	58.95	56.25	53.39	47.06	44.53
	74	57.34	54.71	51.94	45.78	43.32
	76	55.81	53.26	50.56	44.56	42.17
	78	54.37	51.88	49.25	43.41	41.08
	80	53.00	50.57	48.01	42.32	40.04

TABLE—(Continued)

OF SAFE INTERNAL PRESSURES FOR IRON BOILERS.

Birmingham Wire Gauge.		3	4	5	6	7	8
Thickness of Iron.		.259 ¼ Full.	.238 ¼ Scant.	.220 $\frac{7}{32}$	.203 $\frac{6}{32}$ Full.	.180 $\frac{6}{32}$ Scant	.165 $\frac{5}{32}$ Full.
	Dia. In.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.
External Diameter.	24	123.53	113.31	104.58	96.36	85.28	78.07
	26	113.84	104.44	96.40	88.83	78.63	71.99
	28	105.55	96.85	89.40	82.39	72.94	66.79
	30	98.39	90.29	83.36	76.83	68.02	62.29
	32	92.14	84.56	78.07	71.96	63.72	58.35
	34	86.64	79.51	73.42	67.68	59.93	54.89
	36	81.75	75.04	69.29	63.88	56.57	51.81
	38	77.39	71.04	65.60	60.48	53.56	49.06
	40	73.47	67.44	62.29	57.42	50.86	46.58
	42	69.93	64.19	59.29	54.66	48.41	44.35
	44	66.71	61.24	56.57	52.15	46.20	42.32
Long. Seams, Single Riveted.	46	63.78	58.55	54.08	49.87	44.17	40.46
	48	61.09	56.09	51.81	47.77	42.32	38.77
	50	58.62	53.82	49.72	45.84	40 61	37.21
	52	56.35	51.74	47.79	44.07	39.04	35.77
	54	54.24	49.80	46.00	42.42	37.58	34.43
	56	52.28	48.01	44.35	40.90	36.23	33.20
	58	50.46	46.34	42.81	39.48	34.98	32.04
	60	48.77	44.78	41.37	38.15	33.80	30.97
	62	47.18	43.33	40.03	36.91	32.71	29.97
	64	45.69	41.96	38.77	35.75	31.68	29.02
	66	44.30	40.68	37.58	34.66	30.71	28.14
	68	42.99	39.48	36.47	33.64	29.80	27.31
	70	41.75	38.34	35.42	32.67	28.95	26.53
	72	40.58	37.27	34.43	31.76	28.14	25.78
	74	39.48	36.25	33.50	30.89	27.38	25.08
	76	38.43	35.29	32.61	30.08	26.65	24.42
	78	37.44	34.38	31.77	29.30	25.96	23.79
	80	36.49	33.52	30.97	28.56	25.31	23.20

TABLE E—(Continued)

OF SAFE INTERNAL PRESSURES FOR IRON BOILERS.

Birmingham Wire Gauge.		3/8	00	0	1	2
Thickness of Iron.		.375 3/8	.358 3/8 Scant.	.340 11/32	.300 5/16	.284 9/32
	Dia. In.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.
External Diameter.	24	225.81	215.26	204.12	179.49	169.67
	26	207.93	198.23	187.91	165.35	156.34
	28	192.66	183.70	174.23	153.28	144.94
	30	179.49	171.15	162.35	142.86	135.09
	32	168.00	160.21	151.98	133.76	126.49
	34	157.89	150.58	142.86	125.75	118.93
	36	148.94	142.05	134.77	118.64	112.21
	38	140.94	134.43	127.55	112.30	106.22
	40	133.76	127.58	121.06	106.60	100.83
Longitudinal Seams, Double Riveted, Curvilinear Seams, Single Riveted.	42	127.27	121.40	115.20	101.45	95.96
	44	121.39	115.79	109.88	96.77	91.55
	46	116.02	110.68	105.03	92.51	87.52
	48	111.11	106.00	100.59	88.61	83.83
	50	106.19	101.70	96.51	85.02	80.43
	52	102.44	97.73	92.75	81.71	77.33
	54	98.59	94.10	89.27	78.69	74.41
	56	95.02	90.66	86.04	75.81	71.73
	58	91.70	87.49	83.04	73.17	69.23
	60	88.61	84.54	80.24	70.71	66.90
	62	85.71	81.78	77.63	68.40	64.72
	64	83.00	79.17	75.17	66.25	62.68
	66	80.46	76.78	72.87	64.22	60.77
	68	78.07	74.47	70.71	62.31	58.96
	70	75.81	72.34	68.67	60.52	57.26
	72	73.68	70.31	66.74	58.82	55.66
	74	71.67	68.39	64.92	57.22	54.15
	76	69.77	66.60	63.19	55.70	52.77
	78	67.96	64.85	61.56	54.26	51.35
	80	66.25	63.22	60.01	52.90	50.06

TABLE—(*Continued*)

OF SAFE INTERNAL PRESSURES FOR IRON BOILERS.

Birmingham Wire Gauge.		3	4	5	6	7	8
Thickness of Iron.		.259 $\frac{1}{4}$ Full.	.238 $\frac{1}{4}$ Scant.	.220 $\frac{7}{32}$	.203 $\frac{6}{32}$ Full.	.180 $\frac{6}{32}$ Scant	.165 $\frac{5}{32}$ Full
	Dia. In.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.
External	24	154.42	141.64	130.73	120.45	106.60	97.59
Diameter.	26	142.30	130.54	120.50	111.04	98.21	89.99
	28	131.94	121.06	111.76	102.99	91.17	83.48
	30	122.99	112.86	104.19	96.03	85.02	77.86
	32	116.32	105.70	97.59	89.95	79.65	72.94
	34	108.30	99.39	91.78	84.60	74.91	68.61
	36	102.19	93.80	86.61	79.84	70.71	64.76
	38	96.74	88.80	82.00	75.60	66.95	61.32
	40	91.84	84.30	77.86	71.78	63.57	58.23
	42	87.41	80.24	74.11	68.33	60.52	55.44
	44	83.39	76.56	70.71	65.19	57.75	52.90
Long.	46	79.72	73.19	67.60	62.33	55.22	50.58
Seams,	48	76.37	70.11	64.76	59.71	52.90	48.46
Double	50	73.28	67.28	62.11	57.31	50.77	46.51
Riveted.	52	70.43	64.67	59.74	55.08	48.80	44.71
Curvil.	54	67.80	62.25	57.51	53.40	46.98	43.04
Seams,	56	65.35	60.01	55.44	51.12	45.29	41.50
Single	58	63.07	57.92	53.51	49.35	43.72	40.06
Riveted.	60	60.96	55.98	51.71	47.69	42.25	38.71
	62	58.98	54.16	50.03	46.14	40.88	37.46
	64	57.12	52.45	48.46	44.69	39.60	36.28
	66	55.37	50.85	46.98	43.33	38.39	35.18
	68	53.73	49.35	45.59	42.05	37.26	34.14
	70	52.19	47.93	44.28	40.84	36.19	33.16
	72	50.73	46.59	43.04	39.70	35.18	32.23
	74	49.35	45.32	41.87	38.62	34.22	31.36
	76	48.04	44.11	40.76	37.60	33.32	30.53
	78	46.80	42.98	39.71	36.63	32.46	29.74
	80	45.62	41.90	38.71	35.71	31.64	28.99

TABLE—(*Continued*)

OF SAFE INTERNAL PRESSURES FOR STEEL BOILERS.

Birmingham Wire Gauge.		$\frac{3}{8}$	00	0	1	2
Thickness of Steel.		.375 $\frac{3}{8}$	.358 $\frac{3}{8}$ Scant.	.340 $\frac{11}{32}$	.300 $\frac{5}{16}$	.284 $\frac{9}{32}$
	Dia. In.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.
External Diameter.	24	289.03	275.52	261.26	229.74	217.19
	26	266.13	253.73	240.31	211.65	200.08
	28	246.66	235.13	223.01	196.20	185.45
	30	229.74	219.00	207.80	182.85	172.99
	32	215.04	205.06	194.15	171.21	161.91
	34	202.10	192.74	182.85	160.95	152.22
	36	190.63	181.82	172.50	151.86	143.23
	38	180.40	172.06	163.25	143.74	135.96
Longitudinal Seams, Single Riveted.	40	171.21	163.30	154.95	136.44	129.06
	42	162.90	155.39	147.45	129.85	122.83
	44	155.37	148.21	140.66	123.87	117.17
	46	148.50	141.66	134.43	118.41	112.01
	48	142.22	135.67	128.75	113.41	107.29
	50	136.44	130.17	123.53	108.82	100.03
	52	131.12	125.09	118.72	104.59	98.95
	54	126.19	120.39	114.26	100.67	95.24
	56	121.62	116.04	110.13	97.03	91.81
	58	117.37	111.99	106.29	93.65	88.61
	60	113.41	108.21	102.71	90.50	85.63
	62	109.71	104.68	99.36	87.55	82.89
	64	106.24	101.37	96.22	84.79	80.23
	66	102.98	98.26	93.27	82.20	77.77
	68	99.92	95.34	90.32	79.76	75.47
	70	97.03	92.59	87.89	77.43	73.29
	72	94.31	89.99	85.42	75.29	71.24
	74	91.74	87.81	83.09	73.24	69.30
	76	89.30	85.21	80.89	71.29	67.46
	78	86.99	83.01	78.79	69.45	65.72
	80	84.79	80.91	76.81	67.70	64.07

TABLE—(*Continued*)

OF SAFE INTERNAL PRESSURES FOR STEEL BOILERS.

BIRMINGHAM WIRE GAUGE.		3	4	5	6	7	8
Thickness of Steel.		.259 $\frac{1}{4}$ Full.	.238 $\frac{1}{4}$ Scant.	.220 $\frac{7}{32}$	.203 $\frac{3}{16}$ Full.	.180 $\frac{3}{16}$ Scant	.165 $\frac{5}{32}$ Full.
	Dia. In.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.
External Diameter.	24	197.63	181.13	167.33	154.18	136.44	124.91
	26	182.13	167.09	154.24	142.13	125.80	115.10
	28	168.88	154.95	143.04	131.83	116.70	106.85
	30	157.42	144.45	133.36	122.92	108.82	99.65
	32	147.42	135.29	124.91	115.14	101.94	93.36
	34	138.60	127.22	117.47	108.28	95.88	87.81
	36	130.80	120.05	110.86	102.20	90.50	82.89
Long.	38	123.82	113.65	104.96	96.76	85.69	78.49
Seams,	40	117.55	107.90	99.65	91.81	81.37	74.53
Single	42	111.40	102.71	94.85	87.45	77.46	70.95
Riveted.	44	106.71	97.99	90.50	83.44	73.91	67.70
	46	102.04	93.68	86.53	79.78	70.67	64.74
	48	97.74	89.74	82.89	76.43	67.70	62.02
	50	93.07	86.11	79.54	73.35	64.97	59.12
	52	90.15	82.77	76.46	70.50	62.45	57.22
	54	86.78	79.68	73.60	67.87	60.13	55.09
	56	83.65	76.09	70.95	65.43	57.97	53.11
	58	80.74	74.14	68.49	63.16	55.96	51.27
	60	78.02	71.62	66.19	61.07	54.04	49.55
	62	75.49	69.32	64.04	59.06	52.32	47.94
	64	73.11	67.13	62.02	57.20	50.68	46.43
	66	70.88	65.09	60.13	55.45	49.14	45.02
	68	68.77	63.16	58.35	53.52	47.68	43.69
	70	66.79	61.28	56.67	52.27	46.31	42.44
	72	64.92	59.76	55.09	50.81	45.02	41.25
	74	63.16	58.00	53.59	49.43	43.80	40.13
	76	61.48	56.47	52.17	48.12	42.64	39.07
	78	59.90	55.01	50.83	46.88	41.54	38.06
	80	58.39	53.63	49.55	45.65	40.50	37.11

TABLE—(*Continued*)

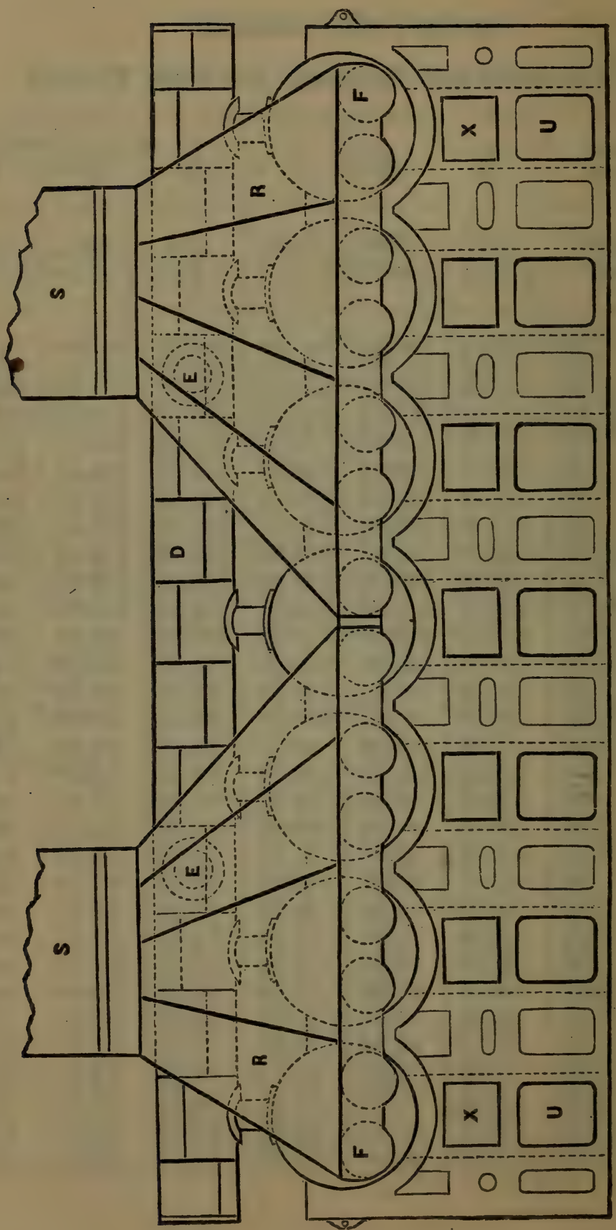
OF SAFE INTERNAL PRESSURES FOR STEEL BOILERS.

Birmingham Wire Gauge.		3/8	00	0	1	2
Thickness of Steel.		.375 3/8	.358 3/8 Scant.	.340 11/32	.300 5/16	.284 9/32
	Dia. In.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.
External Diameter.	24	361.29	344.40	326.58	287.23	271.49
	26	332.67	317.24	300.78	264.56	250.14
	28	308.25	293.91	278.77	237.95	231.90
	30	287.18	273.48	259.75	228.57	216.14
	32	268.80	256.34	243.16	214.01	202.39
	34	252.63	240.93	228.57	201.19	190.28
	36	238.24	227.27	215.62	189.83	179.54
Longitudinal Seams, Double Riveted.	38	225.50	215.08	204.07	179.67	169.95
	40	214.01	204.13	193.69	170.55	161.28
	42	203.63	194.24	184.31	162.31	153.54
	44	194.21	185.26	175.80	154.83	146.47
Curvilinear Seams, Single Riveted.	46	181.21	177.08	168.04	148.01	140.02
	48	177.77	169.55	160.94	141.77	134.12
	50	170.55	162.71	154.41	136.03	128.69
	52	163.90	156.40	148.01	130.73	123.68
	54	157.74	150.49	142.83	125.84	119.05
	56	152.03	145.05	137.61	121.29	114.76
	58	146.72	139.99	132.86	117.01	110.76
	60	141.77	135.26	128.38	113.13	107.03
	62	137.14	130.85	124.20	109.44	103.55
	64	132.80	126.74	120.27	105.99	100.29
	66	128.73	122.83	116.53	102.75	97.22
	68	124.90	119.18	113.13	99.70	94.34
	70	121.29	115.74	109.86	96.85	91.62
	72	117.89	112.49	106.78	94.11	89.05
	74	114.67	109.42	103.87	91.55	86.63
	76	111.62	106.51	101.11	89.12	84.33
	78	108.73	103.76	98.49	86.72	82.15
	80	105.99	101.14	96.01	84.63	80.08

TABLE—(*Concluded*)

OF SAFE INTERNAL PRESSURES FOR STEEL BOILERS.

BIRMINGHAM WIRE GAUGE.		3	4	5	6	7	8
Thickness of Steel.		.259 $\frac{1}{4}$ Full.	.238 $\frac{1}{4}$ Scant.	.220 $\frac{7}{32}$	.203 $\frac{6}{32}$ Full.	.180 $\frac{6}{32}$ Scant	.165 $\frac{5}{32}$ Full.
	Dia. In.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.	lbs. per sq. in.
External	24	247.06	226.62	209.16	192.72	175.63	156.14
Diameter.	26	227.67	208.87	192.80	177.66	157.25	143.98
	28	211.10	193.69	178.80	164.78	145.87	133.57
	30	196.78	180.57	166.71	153.65	136.03	124.57
	32	184.28	169.75	156.14	143.92	127.43	116.70
Long.	34	173.27	159.06	146.84	135.35	119.85	109.77
Seams,	36	163.50	150.07	138.58	127.75	113.13	103.61
Single	38	154.73	142.07	131.20	120.95	107.12	98.11
Riveted.	40	146.94	134.88	124.57	114.84	101.71	93.16
Curvil.	42	139.85	128.38	118.57	109.32	96.82	88.69
Seams,	44	133.42	122.48	113.13	104.30	92.39	84.64
Single	46	127.55	117.10	108.16	99.73	88.34	80.92
Riveted.	48	122.18	112.17	103.61	95.54	84.63	77.53
	50	117.24	107.64	99.43	91.68	81.22	74.41
	52	112.69	103.43	95.53	88.13	78.07	71.53
	54	108.47	99.60	92.00	84.84	75.16	68.86
	56	104.56	96.01	88.69	81.79	72.46	66.39
	58	100.92	92.67	85.61	78.95	69.95	64.08
	60	97.53	89.56	82.74	76.26	67.60	61.60
	62	94.36	86.65	80.11	73.17	65.44	59.93
	64	91.38	83.98	77.53	71.52	63.35	58.04
	66	88.59	81.36	75.16	69.32	61.42	56.28
	68	85.97	78.95	72.94	67.23	59.60	54.61
	70	83.49	76.68	70.84	65.34	57.89	53.05
	72	81.16	74.53	68.86	63.51	56.28	51.56
	74	78.95	72.50	66.72	61.78	54.75	50.16
	76	76.86	70.58	65.21	60.15	53.30	48.84
	78	74.87	68.76	63.52	58.60	51.93	47.58
	80	72.99	66.96	61.94	57.12	50.62	46.39



MARINE FLUE BOILER.

U U, ash-pit doors. X X, fire-doors. F F, flues. R R, bonnets. E E, steam-pipes. D, steam-drum. S S, uptakes.

MARINE BOILERS.

There is now, as there always has been, a great diversity of opinion among engineers in regard to the true principles upon which to design a marine boiler which shall produce the greatest effect with the least stowage, first cost and subsequent labor, and fuel. But experience has shown that the best that can be done is to determine which of these considerations shall have the least weight, and to be governed accordingly, looking, as a guide, to practice rather than any assumed theoretical principles.

For land purposes, there is hardly any limit to the size or weight of a boiler except first cost; it is easy, therefore, to design and construct one with sufficient heating surface, water space, and steam room. But in designing a marine boiler the case is quite different, as the designer is restricted both in room and weight; for if the vessel be occupied or loaded down with boilers, it detracts from the room and capacity that should be devoted to other purposes.

Marine boilers are of necessity either flue or tubular, since the flame must be within the shell of the boiler; but in this arrangement they are almost as various as the makers. The large flue is preferable because less liable to choke with soot, ashes, cinders, or salt which may come from leakage. But in situations which restrict length, height, and width of boiler, the only method of producing in a flue boiler such extent of fire surface as will extract all the heat capable of being used to advantage in generating steam, is to reduce the size and multiply the number of flues.

The most ordinary forms of marine boilers are the horizontal and vertical; and, so far as efficiency is concerned, there does not appear to be any great difference between

them where equal surfaces are presented to the action of the fire; but there are many things, particularly in sea-going steamers, to be considered, and for them that boiler is the best which gives equal effect, occupies the least space, and affords the best facilities for cleaning and repairs.

A certain proportion between the area of the grate and the total heating surface has been found productive of the best results, with a given description of fuel; but any alteration in the quality of the fuel used will be found to affect this result materially. Consequently, no general rule can be laid down for the design of marine boilers that will answer for all kinds of fuel, nor is it at all likely that any one form will ever fulfil all the varied conditions under which such boilers may be placed.

A consideration of great importance in the construction of marine boilers is their capacity to contain water and steam. This, of course, depends upon the size of the boiler and the proportion of space occupied by flues or tubes, as, if the space within it be nearly filled with flues, there can be but little room left for water.

In fixing on the proper capacity of the water-space of a marine boiler, there are not such peculiar difficulties as in the case of the steam-chamber, and any one at a first view of the matter would say, as many do without sufficient consideration, that there cannot be too little water, provided the boiler is filled to the proper height; for it is quite obvious the smaller the quantity of water the less will be the expenditure of the fuel during the first getting up of the steam after each stoppage of the engine. It is, however, not the "getting up" the steam, but the keeping it up, that ought to be considered of most consequence. It is a prevailing opinion that, after the steam is once got up, there is no material difference between keeping a large

quantity of water boiling and a small quantity, provided the escape of heat is prevented by sufficiently clothing the boiler with non-conducting substances; but on this subject engineers differ. Why practical men should differ in opinion on so plain a matter is unaccountable.

The quantity of water carried must exceed that of the evaporation in a given time, in order that the supply of feed-water may not greatly reduce the temperature of the water in the boiler and check the formation of steam. There must in all cases be a sufficient height of water in the boiler to prevent the flues or crown-sheet from becoming bare in case the supply of feed-water be neglected, or the vessel pitches in a rough sea.

Steam room is understood to be the space in the shell of the boiler above the level of the water, and in marine boilers should be from ten to twelve times the capacity of the cylinder of the engine. This proportion has of necessity a very narrow limit of variation, as, if the steam room be less than the above proportions, at every stroke of the engine the pressure of steam on the surface of the water is liable to be reduced to such an extent as to produce violent ebullition or foaming.

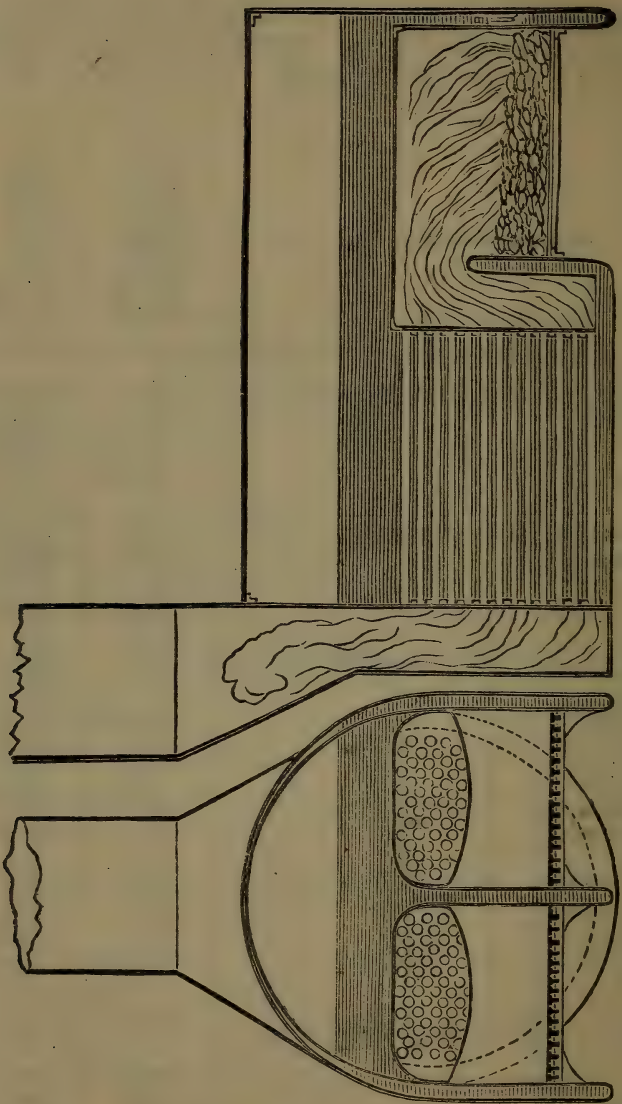
When boilers are so constructed that steam cannot be taken off above the level of the water without the danger of working water into the steam-cylinder, it becomes necessary to resort to the expedient of attaching a steam-dome to the boiler. This steam-dome is constructed either inside or around the smoke-pipe, which, though not adding much to the cubic capacity of the steam room, has the effect of superheating the steam, or imparting to it an extra heat, which greatly increases its expansive force and renders it less liable to condense in the passages between the boiler and the cylinder.

PROPORTIONS OF HEATING SURFACE TO CYLINDER AND GRATE SURFACE OF NOTED OCEAN, RIVER, AND FERRY-BOAT STEAMERS.

NAME OF STEAMER.		Number of sq. feet of heating surface to 1 cubic foot of cylinder.	Number of sq. feet of heating surface to 1 sq. ft. of grate surface.
Powhatan,	U. S. N	14.8	22.3
Susquehanna,	" " "	16.25	25.
Mississippi,	" " "	12.6	18.6
San Jacinto,	" " "	17.75	27.
Saranac,	" " "	14.5	27.25
Princeton,	" " "	28.8	22.
Michigan,	" " "	15.	19.75
Vixen,	" " "	18.	16.
Massachusetts,	" " "	77.4	33.6
Georgia,	Merchant Steamer......	13.	22.25
Washington,	" "	12.	23.5
United States,	" "	8.2	21.9
Northerner,	" "	12.75	24.9
Falcon,	" "	12.75	20.8
Philadelphia,	" "	15.	21.
Republic,	" "	20.	31.
Ohio,	" "	13.	22.25
Hermann,	" "	14.8	30.6
Cherokee,	" "	12.17	23.7
Union	" "	118.	66.4
Constitution,	" "	93.	34.5
Golden Gate,	" "	17.	32.8
Monumental City,	" "	51.	31.5
El Dorado,	" "	14.	26.8
City of Pittsburg,	" "	30.9	35.5
Pioneer,	" "	28.	33.5
Albatross,	" "	57.25	32.75
Osprey,	" "	27.5	34.
Humboldt,	" "	12.8	18.6
Franklin,	" "	11.3	28.4
Arctic,	" "	21.5	33.25
Baltic,	" "	21.5	33.25
Pacific,	" "	21.5	33.25
Atlantic,	" "	21.5	33.25

PROPORTIONS OF HEATING SURFACE TO CYLINDER AND GRATE SURFACE OF NOTED OCEAN, RIVER, AND FERRY-BOAT STEAMERS.

NAME OF STEAMER.	Number of sq. feet of heating surface to 1 cubic foot of cylinder.	Number of sq. feet of heating surface to 1 sq. ft. of grate surface.
May Flower, Merchant Steamer.........	15.4	31.71
Empire State, " "	14.	24.5
America, " "	33.75	32.25
Knoxville, " "	15.33	63.1
North America, River Steamer.............	13.6	22.3
South America, " "	14.15	24.9
Oregon, " "	12.	31.3
Alida, " "	13.6	27.9
Niagara, " "	12.5	27.
Joseph Belknap, " "	21.	27.5
Mountaineer, " "	12.	32.
New World, " "	11.3	25.17
Traveller, " "	12.5	21.3
Isaac Newton " "	10.6	28.2
Roger Williams, " "	12.	19.2
Thomas Powell, " "	16.25	25.5
Armenia, " "	16.	24.5
America, " "	21.7	26.
Bay State, " "	12.	29.3
Empire State, " "	11.	25.
Baltimore, " "	22.10	42.37
J. M. White, Western river Steamer........	30.	26.
Rescue, Steam-tug......................	63.3	28.
Anglo-Saxon, "	10.1	23.
Merchant, Ferry-boat.........................	25.5	38.
Seneca, "	16.	20.
Onalaska "	15.5	19.6
John Fitch, "	17.25	22.3
Average...............	29.83	28.08



Horizontal Tubular Marine Boiler.

SETTING MARINE BOILERS.

In steamships, it is advisable to place the boilers at or as near the centre of the ship as possible, and at equal distances from the keelsons on each side, in order that there may be no difficulty in keeping the ship in trim, or changing her running line when necessary.

The boiler foundations ought to be laid so as to make the lower face of the boilers and the line of the tubes or flues parallel with the load line fore and aft. They should then be firmly secured with braces bolted to the hull, in order to prevent the possibility of their being displaced by any strain or motion to which the ship may be subjected.

BEDDING MARINE BOILERS.

The manner of bedding marine boilers is a point of much importance, and will materially affect the durability of the bottom plates. The general practice in this country is to form a close platform of 2½ or 3 inch yellow pine plank over the keelsons, upon which the boiler is imbedded with a cement composed of drying oil and whiting, laid about 1½ inches over the plank; this cement sets quite hard, and prevents any dampness or bilge water from rusting or corroding the boiler. The cement is also intended to stop any leaks that may break out in the bottom or water-legs.

Unfortunately, however, the benefits to be derived from such a bedding are frequently lost in consequence of the unequal degrees of expansion between the cement and the boiler. The best practice is, perhaps, that of resting the boiler on saddles of cast-iron fixed on the boiler bearers, which leaves the bottom exposed for examination, painting, and small repairs, if necessary; the bottom of the vessel under the boilers being at the same time kept clean and dry by the bilge-pumps.

CLOTHING OF MARINE BOILERS.

Although it must be allowed that in all cases the clothing of marine boilers with non-conducting substances, such as hair-felt, wood, etc., is highly advantageous for the pro-

duction of steam, yet this practice is alleged in some instances to have induced a rapid wear in the plates of the boiler. This unlooked-for result is most apparent in boilers which are frequently used and disused alternately, the corrosion taking place on the interior surface.

The conjecture as to its cause is, that owing to the alternate wetting and drying of the plates of the clothed boiler, the rust may be more apt to scale off, and thus constantly present a clean surface for corrosion, this action recurring each time that the water is blown out of the boiler; but when, on the other hand, the boiler is naked, the internal surface never thoroughly dries, owing to the evaporation being checked by the low temperature, and the saturation of the confined air.

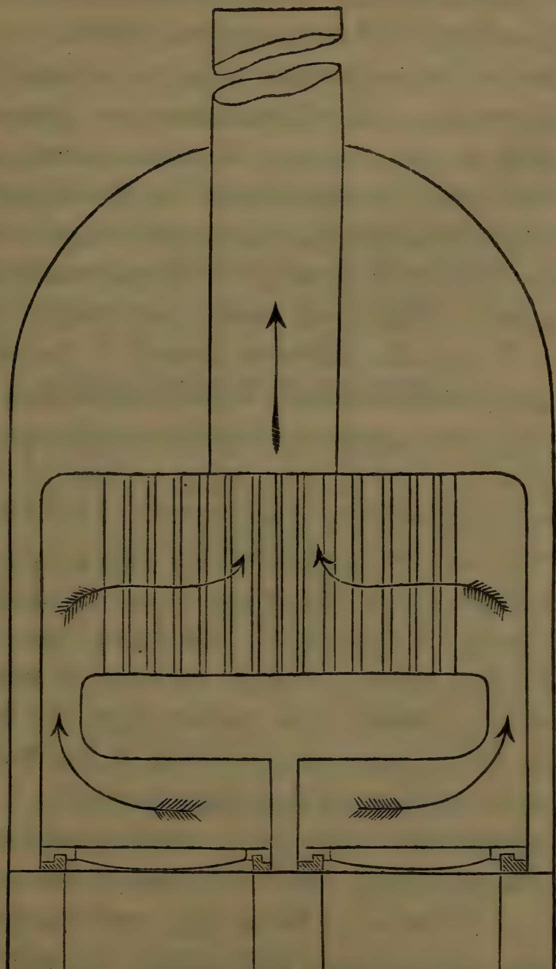
The clothing of marine boilers which make long voyages is never attended with these injurious results, nor are they ever experienced in land boilers.

CARE OF MARINE BOILERS.

Marine boilers require much attention, both at sea and in port, especially if they be complex tubular constructions. The great points at sea are, the firing, the feeding, and the blowing off; the great points in port, the cleaning and repairing. If the boiler be blown off by means of blow-off cocks, the operation should be performed twice in the watch, or once in every two hours. The feed should be so set that the water will rise in the course of two hours from a little below the middle to near the top of the glass gauge tube; the rule being to blow off so frequently, or so much, as to prevent any accumulation of scale within the boiler.

In boilers furnished with brine pumps, reliance must not be placed upon the pumps always acting well, and once every watch some water should be drawn off from

the boiler, to be tested by a salt gauge, to see whether it is too salt or not. When the water has been evaporated to such an extent as to reduce its volume some three- or four-fold, the saltness becomes so excessive that solid salt is liable to form upon the exposed surfaces.



Vertical Tubular Marine Boiler.

The saltness of ordinary sea-water varies somewhat in different places, but, as a general rule, there is about one pound of salt in every thirty-three of sea-water. When,

by boiling, the proportion of salt is increased to about $\frac{4}{33}$, the formation of a scale consisting mainly of salt is likely to commence. It is important, therefore, to blow out a portion and to supply its place by new, so often as to keep the water fresher than $\frac{4}{33}$; but, on the other hand, every exchange of hot water for cold diminishes the supply of steam and increases the consumption of fuel.

To ascertain the saltness of water as accurately as possible, hydrometers and salinometers are generally employed. But in the absence of these instruments, the engineer may make one for himself in the following manner: Take a glass phial or cologne bottle, pour into it so much shot that it will nearly sink in sea-water, and then cork it tightly. Take any convenient weight of boiling water, say 33 pounds; dissolve therein 1 pound of salt, and then put the phial into it, turned upside down, so that the shot will rest against the cork; make a mark with a file at the point at which the water stands on the phial; this represents the saltness of sea-water. Then add another pound of salt to the water, marking the point, as before, on the phial at which the water stands, and repeat the operation until 12 pounds of salt have been added, at which point the water will have received as much salt as it can dissolve; transfer the marks upon the bottle to a paper scale, which paste on the inside of the bottle in exactly the same position as the original marks.

The engineer will then have a salt gauge which will tell the saltness of brine from the point of sea-water up to the point of situation. Reckoning sea-water at 1, the water in the boiler should not exceed the saltness represented by 4, at which point the water contains $\frac{4}{33}$ of salt. It is not probable that this rude contrivance will often be made, but the description may be of service in

explaining the nature of the more elaborate and accurate instruments sold for the purpose.

If a vessel is to remain in port any length of time after the boilers become cool, the hand-hole plates over the furnaces should be removed in order to ascertain if there are any heavy deposits on the crown-sheet; the bottom hand-hole plates should also be taken out, so as to allow the water to drain out and permit a current of air to pass through, as mere dampness is more injurious than actual use, in consequence of the rapid oxidation it induces. In cases where circumstances forbid the removal of the bottom hand-hole plates and draining of the boilers, it is preferable that they should remain full rather than have only a small quantity of water in the bottom.

REPAIRING STEAM-BOILERS.

Repairing steam-boilers is generally attended with more or less difficulty, arising from the peculiar type of boiler and the cramped and inconvenient location in which the repairs generally have to be made, and also, in many cases, the want of proper facilities at the time and location.

The two most ordinary methods of patching boilers when they crack or burn out, and, in fact, the only two that can be successfully employed, are the hard and soft patching. The former is the most permanent and reliable; but it is only practicable where nearly all the facilities required by boiler-makers are at hand. The soft patching process is generally resorted to in repairing the boilers of steam-vessels when at sea, or the boilers of locomotives in sections of country where it is impossible to employ the hard patch.

To apply the hard patch, it is necessary to lay off its dimensions on the plate to be patched, allowing from 1 to $1\frac{1}{4}$ inches of sound material outside of the crack or flaw.

The patch is next drilled for the rivet-holes, and its edges chipped; it is then placed over the defective part, and the holes marked through it by means of a small tube dipped in white paint. The holes are next drilled in the sheet with a ratchet-drill and brace; the defective material in the sheet is then cut out, in order to allow the water to come in contact with the patch, which is then riveted on and calked; but in case it should be found impossible, for want of space, to rivet it, it is put on with tap-bolts.

The soft patch is prepared in the same way, but applied differently, it being first covered with a heavy coat of cement, and then attached to the defective place by means of bolts, nuts, and washers; a grummet of hemp being placed under the head of each bolt and washer to make it steam- and water-tight.

The hard patch is more suitable for furnaces and parts of the boiler exposed to the action of the fire, while the soft answers very well for the steam room and water space.

TUBES.

The use of tubes is to conduct heat to the surrounding water at the least possible cost, the items of cost being, 1st, waste heat; 2d, maintenance of tubes. Granted that the best conducting tube is the least durable, and that the poorest conducting tube is the most durable, the question is, By avoiding which species of expense shall the highest economy be attained?

The resistance of tubes is manifestly due entirely to their hardness; the materials ranging in the following order: steel, iron, brass, copper.

Iron has, heretofore, more especially where anthracite coal has been used as fuel, nearly superseded all other materials for tubing on account of its hardness and good flanging qualities; but at the present time steel seems to

afford better results than any other material, as the tubes can be made lighter, and possess steaming qualities equal, if not superior, to either copper or brass; while the nature of the material affords the requisite degree of surface resistance to the chemical action of the water in the boiler.

The failure of tubes might in the majority of cases be attributed to a contracted water space, bad circulation between them, and the deposit of scale adhering to the outer surface caused by impurities in the water.

Diameter and Arrangement of Tubes. — Tubes two inches in diameter, placed in vertical rows $\frac{3}{4}$ of an inch apart, give most satisfactory results, as such an arrangement admits of an easy circulation of the water and free escape of steam from the heating surface to the steam-dome, besides giving ready access to the mud in its passage from the water to the bottom of the boiler.

Crowding tubes in tubular boilers is often carried to an extreme with the view of getting more surface, but without regarding the other conditions of steam-raising. Heating surface in the abstract is one thing, its efficiency is another, as the under portions of the tubes and internal flues are almost worthless for steam-raising, not only on account of the difficulty the steam has in escaping from the surface on one side, but also in consequence of the deposit of soot, ashes, and flue dirt which is the rule on the other.

The incrustation also accumulates much more rapidly, and to a greater thickness, on the under side than on the crown of tubes, especially of large diameter, principally on account of the comparatively quiescent state of the water in contact with the former.

Assuming the gases entering a tube to be all of the same temperature, the particles striking against the upper sur-

face must give up part of their heat, and, in cooling, descend by virtue of their increased gravity, despite the onward and upward force due to the momentum of the mass which opposes their descent. The hot particles immediately behind and beneath these will come in contact with the upper surface a little further on, and so a species of convection is kept up as the gases sweep along.

In horizontal tubes various means have been devised for extracting more of the heat out of the gases than they will yield by radiation or conduction through their mass, by breaking the current at intervals, and so bringing fresh portions of the gases in contact with the plates, by giving them a zigzag motion; this, however, has the effect of impairing the draught and, in most cases, of causing a reduction in the evaporative capacity of the boiler.

In passing up through vertical tubes gases act at a disadvantage for imparting their heat to the plates. The particles cooled by contact with the sides on entering have no tendency to make way for those in the middle of the current that still retain their heat, which can therefore only be indifferently imparted by radiation or conduction.

The evaporative efficiency of tubes, as before stated, depends on the nature, condition, and thickness of the material forming the tubes. In tubes manufactured from homogeneous metal, the resistance to internal conduction is proportional directly to the distance the heat has to traverse or to the thickness of the tube, and inversely to the difference of temperatures between the two surfaces.

TABLE

OF SUPERFICIAL AREAS OF EXTERNAL SURFACES OF TUBES OF VARIOUS LENGTHS AND DIAMETERS IN SQUARE FEET.

These tables are designed to facilitate the calculation of the heating surface of the tubes in tubular boilers, and are adapted for tubes of various lengths, from 8 to 13 feet, advancing by inches, and of various diameters, from $1\frac{5}{8}$ to $2\frac{1}{4}$ inches, advancing by $\frac{1}{8}$ of an inch.

EXPLANATION.

The large figures at the end of the horizontal lines give the length of the tubes in feet, and the small intermediate figures on the same line give the additional inches. The vertical column on the left gives the diameter of the tubes in inches. The numbers in the tables represent the superficial area of one tube in square feet, and decimal parts thereof, for the different lengths and diameters of tubes required.

EXAMPLE.

Required the heating surface of 163 tubes, $1\frac{3}{4}$ inches diameter and 11 feet 10 inches long. Thus, having found the length (11 feet 10 inches) in the above named horizontal line of figures, trace downwards to the line opposite the diameter ($1\frac{3}{4}$) in the vertical column on the left, where will be found the number 5·421, being the area of the tube, and which, being multiplied by the number of tubes (163), gives the total area of 883,623 square feet, thus reducing the whole process to a simple matter of multiplication.

35 *

SUPERFICIAL AREAS OF EXTERNAL SURFACES OF TUBES OF VARIOUS LENGTHS AND DIAMETERS IN SQUARE FEET.

DIAMETER OF TUBE. Inches.	FEET. 8	INCHES. 1	2	3	4	5	6	7	8	9	10	11
1⅝	3.403	3.438	3.474	3.509	3.545	3.580	3.616	3.651	3.686	3.722	3.757	3.793
1¾	3.665	3.703	3.741	3.779	3.817	3.856	3.894	3.932	3.970	4.008	4.046	4.085
1⅞	3.926	3.967	4.008	4.094	4.090	4.131	4.172	4.213	4.254	4.295	4.335	4.376
2	4.188	4.232	4.276	4.319	4.363	4.406	4.450	4.494	4.537	4.581	4.624	4.668
2⅛	4.450	4.496	4.543	4.589	4.636	4.682	4.728	4 775	4.821	4.867	4.914	4.960
2¼	4.712	4.761	4.810	4.859	4.908	4.957	5.006	5.055	5.105	5.154	5.203	5.252
Inches.	9 ft.	1 in.	2 in.	3 in.	4 in.	5 in.	6 in.	7 in.	8 in.	9 in.	10 in.	11 in.
1⅝	3.828	3.864	3.899	3.935	3.970	4.006	4.041	4.076	4.112	4.147	4.183	4.218
1¾	4.123	4.161	4.199	4.237	4.276	4.314	4.352	4.390	4.428	4.466	4.505	4.543
1⅞	4.417	4.458	4.499	4.540	4.581	4.622	4.663	4.704	4.745	4.785	4.826	4.867
2	4.712	4.756	4.799	4.843	4.886	4.930	4.974	5.017	5.061	5.104	5.148	5.192
2⅛	5.006	5.053	5.099	5.145	5.192	5.238	5.285	5.331	5.377	5.424	5·470	5.516
2¼	5.301	5.350	5.399	5.448	5.497	5.546	5.595	5.645	5.694	5.743	5.792	6.841

SUPERFICIAL AREAS OF EXTERNAL SURFACES OF TUBES OF VARIOUS LENGTHS AND DIAMETERS IN SQUARE FEET.

DIAM. OF TUBE.	FEET.	INCHES.										
Inches.	10	1	2	3	4	5	6	7	8	9	10	11
1⅝	4.254	4.289	4.325	4.360	4.396	4.431	4.466	4.502	4.537	4.573	4.608	4.644
1¾	4.581	4.619	4.657	4.696	4.734	4.772	4.810	4.848	4.886	4.924	4.963	5.001
1⅞	4.908	4.949	4.990	5.031	5.072	5.113	5.154	5.195	5.235	5.276	5.317	5.358
2	5.236	5.279	5.323	5.366	5.410	5.454	5.497	5.541	5.584	5.628	5.672	5.715
2⅛	5.563	5.609	5.655	5.702	5.748	5.795	5.841	5.887	5.934	5.980	6.026	6.073
2¼	5.890	5.939	5.988	6.037	6 086	6.135	6.184	6.234	6.283	6.332	6.381	6.430

Inches.	11 ft.	1 in.	2 in.	3 in.	4 in.	5 in.	6 in.	7 in.	8 in.	9 in.	10 in.	11 in.
1⅝	4.679	4.715	4.750	4.785	4.821	4.856	4.892	4.927	4.963	4.998	5.034	5.069
1¾	5.039	5.077	5.115	5.154	5.192	5.230	5.268	5.306	5.345	5.383	5.421	5.459
1⅞	5.399	5.440	5.481	5.522	5 563	5.604	5.644	5.685	5.726	5.767	5.808	5.849
2	5.759	5.803	5.846	5.890	5.934	5.977	6.021	6.064	6.108	6.152	6.195	6.239
2⅛	6.119	6.165	6.212	6.258	6.304	6.351	6.397	6.444	6.490	6.536	6.583	6.629
2¼	6.479	6.528	6.577	6.626	6.675	6.724	6.774	6.823	6.872	6.921	6.970	7.019

SUPERFICIAL AREAS OF EXTERNAL SURFACES OF TUBES OF VARIOUS LENGTHS AND DIAMETERS IN SQUARE FEET.

DIAMETER OF TUBE. Inches.	FEET. 12	INCHES. 1	2	3	4	5	6	7	8	9	10	11
1⅝	5.105	5.140	5.175	5.211	5.246	5.282	5.317	5.353	5.388	5.424	5.459	5.494
1¾	5.497	5.535	5.574	5.612	5.650	5.688	5.726	5.764	5.803	5.841	5.879	5.917
1⅞	5.890	5.931	5.972	6.013	6.054	6.094	6.135	6.176	6.217	6.258	6.299	6.340
2	6.283	6.326	6.370	6.414	6.457	6.501	6.544	6.588	6.632	6.675	6.719	6.762
2⅛	6.675	6.722	6.768	6.814	6.861	6.907	6.954	7.000	7.046	7.093	7.139	7.185
2¼	7.068	7.117	7.166	7.215	7.264	7.314	7.363	7.412	7.461	7.510	7.559	7.608
Inches.	13 ft.	1 in.	2 in.	3 in.	4 in.	5 in.	6 in.	7 in.	8 in.	9 in.	10 in.	11 in.
1⅝	5.530	5.565	5.601	5.636	5.672	5.707	5.743	5.778	5.814	5.849	5.884	5.920
1¾	5.955	5.994	6.032	6.070	6.108	6.146	6.184	6.223	6 261	6.299	6.337	6.375
1⅞	6 381	6.422	6.463	6.504	6.544	6.585	6.626	6.667	6.708	6.749	6 790	6.831
2	6.806	5.850	6.894	6.937	6.981	7 024	7.068	7.112	7.155	7.199	7.242	7·286
2⅛	7.232	7.278	7.324	7.371	7.417	7.463	7.510	7.556	7.603	7.649	7.695	7.742
2¼	7.657	7.706	7.755	7.804	7.853	7.903	7.952	8.001	8.050	8.099	8.148	8.197

BOILER FLUES.

The well established law that the strength of cylinders is inversely as their diameters, and the hitherto undisputed axiom among practical engineers, that cylindrical tubes or boiler flues when subjected to uniform external pressure were equally strong in every part regardless of length, led to erroneous opinions regarding the strength of boiler flues.

For flues to collapse, under the ordinary working pressure of steam in what was supposed to be properly proportioned and well-made boilers, was formerly not an unusual occurrence; and although many theories were advanced on the subject, it was not until the celebrated English engineer, William Fairbairn, LL.D., F. R. S., made an extensive set of experiments on the strengths of tubes of various forms, sizes, and lengths, that the hidden weakness was revealed.

These experiments were made by hydrostatic pressure, applied both externally and internally, to test the strength under ordinary conditions of practice, and they proved conclusively that the strength of flues exposed to external pressure, as ordinarily used, is *inversely as the length;* that is, a flue 20 feet long will collapse with just half the pressure of a flue 10 feet long, everything else being equal; in other words, a flue 20 feet long, which would bear a pressure of 90 pounds per square inch, if shortened to 10 feet, or, what is the same thing in effect, if it be hooped in the middle of its length by angle or T iron, it will then bear a pressure of 180 pounds per square inch.

Although it had long been established that a circle is the strongest possible form that can be made, and that no deviation from it can be made without reduction of strength; yet it was not previously known that a 9-inch diameter

of tube was reduced in strength more than one-third by deviating from a circle only sufficient to make a lap-joint, the ratio being as 7 to 10 — so proved by tests.

When pressure is exerted within a tube or cylinder with spherical ends, the tube can only give way by the metal being torn asunder; and the tendency of the strain is to cause the tube to assume the true cylindrical figure, or spherical form—the form of greatest resistance. With pressure exerted on the *outside* of a tube, the tendency of that pressure is to crush in the tube — to flatten it.

It is a well-known fact that iron of any strength, when formed into a tube, will bear a much greater strain to tear it asunder, if that pressure be applied *internally*, than it will bear without crushing in, when applied *externally*. A bar of iron, when used as a tie-rod, will resist a very large amount of tearing force; but that same bar, placed as a prop only, under the weight exerted in the former case, would be doubled up and crushed out of form.

The inner tubes of boilers are nothing more nor less than a series of props, as they have to sustain the immense weight of the pressure exerted externally on their diameter. The constant and never-ceasing tendency is for those props to give way — for the cylindrical tube to depart from the form of greatest resistance, to become flattened or bulged, and ultimately crushed in.

The foregoing conclusions show the imperative necessity of adhering to the true circle for boiler flues, more especially where high-pressure steam is used.

Rule *for finding the Safe External Pressure on Boiler Flues.* — Multiply the square of the thickness of the iron by the constant whole number 806,300; divide this product by the diameter of the flue in inches; divide the quotient by the length of the flue in feet; divide this quotient by 3. The result will be the safe working pressure.

EXAMPLE.

Diameter, 13 inches. Length, 10 feet.

Thickness, $\frac{3}{8}$ of an inch.

$\frac{3}{8} \times \frac{3}{8} = \frac{9}{64}$.

$$\begin{array}{rl} 13 & \text{diameter.} \\ 10 & \text{length.} \\ \hline 130 & \\ 3 & \\ \hline 390 & \end{array}$$

$\frac{9}{64} \times 806{,}300 = \frac{7256700}{64} \div 390 = \frac{7256700}{24960} = 290{\cdot}73$ safe external pressure.

TABLE

OF SQUARES OF THICKNESSES OF IRON, AND CONSTANT NUMBERS TO BE USED IN FINDING THE SAFE EXTERNAL PRESSURE FOR BOILER FLUES.

Birmingham Gauge.							
$\frac{3}{8}$	·375	×	·375	×	806,300	=	113,385·937500
00	·358	×	·358	×	806,300	=	103,338·633200
0	·340	×	·340	×	806,300	=	93,208·280000
1	·300	×	·300	×	806,300	=	72,567·000000
2	·284	×	·284	×	806,300	=	65,032·932800
3	·259	×	·259	×	806,300	=	54,087 410300
4	·238	×	·238	×	806,300	=	45,672 057200
5	·220	×	·220	×	806,300	=	39,024·920000
6	·203	×	·203	×	806,300	=	33,226·816700
7	·180	×	·180	×	806,300	=	26,124·120000
8	·165	×	·165	×	806,300	=	21,951·517500

Explanation.—The column on the left-hand side of the page, $\frac{3}{8}$, 00, 0, 1, etc., represents the number of the boiler iron according to the Birmingham wire gauge; the second and third columns, ·375, ·358, etc., represent the decimal parts of an inch, the inch being taken as 10,000, which columns being multiplied together give the square of the thickness of the iron; the fourth column represents the constant number 806, 300, by which we multiply the several squares of the thicknesses; the fifth column represents the several products.

TABLE

OF SAFE WORKING EXTERNAL PRESSURES ON FLUES 10 FEET LONG.

BIRMINGHAM GAUGE.	$\frac{3}{8}$	00	0	1	2
Thickness of Iron.	.375	.358	.340	.300	.284
Diam. in In.					
6	629.92	574.10	517.82	403.15	361.29
7	539.93	492.08	443.85	345.56	313.96
8	472.44	430.58	388.37	302.36	270.97
9	419.95	382.74	345.22	273.95	240.86
10	377.95	344.46	310.69	241.89	216.78
11	343.59	313.15	282.45	219.56	199.80
12	314.12	287.05	258.91	201.58	180.65
13	290.73	264.97	238.99	186.07	166.75
14	269.97	246.04	221.92	172.78	154.84
15	251.97	229.64	207.13	161.26	144.51
16	236.22	215.28	194.18	151.18	135.49
17	222.33	202.62	182.76	142.28	125.06
18	209.97	191.18	172.61	134.38	120.43
19	198.92	181.12	163.52	127.72	114.09
20	188.98	172.23	155.35	120.95	108.39
21	179.98	164.02	147.95	115.19	103.23
22	170.28	156.57	141.23	109.95	98.53
23	164.33	149.76	135.08	105.17	94.25
24	157.48	143.53	129.46	100.79	90.32
25	151.18	137.78	124.28	96.76	86.71
26	145.37	132.79	119.50	93.03	83.37
27	139.98	127.58	115.07	89.58	80.28
28	134.98	123.02	110.96	86.39	77.42
29	130.33	118.79	107.14	83.41	74.75
30	125.98	114.82	103.56	80.63	72.25
32	118.11	107.65	97.09	75.55	67.74
34	111.16	101.31	91.38	71.14	63.75
36	104.99	95.68	86.30	67.19	60.21
38	99.46	90.65	81.76	63.65	57.04
40	94.49	86.11	77.67	60.47	54.19
42	89.99	82.00	73.97	57.59	51.61

TABLE—(*Continued*)

OF SAFE WORKING EXTERNAL PRESSURES ON FLUES 10 FEET LONG.

BIRMINGHAM GAUGE.	3	4	5	6	7	8
Thickness of Iron.	.259	.238	.220	.203	.180	.165
Diam. in In.						
6	300.49	253.73	216.81	184.59	145.14	121.95
7	257.56	217.49	185.83	158.22	124.40	104.53
8	225.36	190.30	162.60	138.45	108.85	91.46
9	200.32	169.16	140.83	123.06	96.76	81.30
10	180.29	152.24	130.08	110.76	87.08	73.17
11	163.90	138.40	118.26	100.69	79.16	66.51
12	150.24	126.87	108.40	92.30	72.56	60.97
13	138.69	117.11	100.06	85.20	66.98	56.28
14	128.78	108.74	92.92	79.11	62.20	52.26
15	120.19	101.49	86.72	73.83	58.05	48.78
16	112.68	95.15	81.30	69.22	54.42	46.10
17	106.05	89.55	76.51	65.15	51.22	43.04
18	100.16	84.58	72.26	61.53	48.37	40.65
19	94.89	80.13	68.46	58.29	45.83	38.51
20	90.15	76.12	65.04	55.37	43.54	36.58
21	85.85	72.49	61.92	52.74	41.46	34.84
22	81.95	69.20	59.12	50.34	39.58	33.25
23	78.38	66.19	56.55	48.15	37.86	31.81
24	75.12	63.43	54.20	46.14	36.28	30.48
25	72.11	60.89	52.11	44.30	34.83	29.26
26	69.34	58.55	50.03	42.59	33.49	28.91
27	66.77	56.38	48.17	41.02	32.25	27.10
28	64.38	54.37	46.45	39.55	31.10	26.13
29	62.16	52.49	44.85	38.19	30.02	25.23
30	60.09	50.74	43.36	36.91	29.02	24.39
32	56.34	47.57	40.65	34.61	27.21	22.86
34	53.02	44.77	38.25	32.57	25.61	21.52
36	50.08	42.38	36.13	30.76	24.18	20.32
38	47.44	40.06	34.23	29.14	22.91	19.25
40	45.07	38.06	32.52	27.68	21.77	18.29
42	42.13	36.24	30.97	26.37	20.73	17.42

TABLE—(*Continued*)

OF SAFE WORKING EXTERNAL PRESSURES ON FLUES 20 FEET LONG.

BIRMINGHAM GAUGE.	$\frac{3}{8}$	00	0	1	2
Thickness of Iron.	.375	.358	.340	.300	.284
Diam. in In.					
6	314.96	287.05	258.91	201.58	180.65
7	269.97	246.04	221.93	172.78	156.98
8	236.22	215.29	194.18	151.18	135.49
9	209.97	191.37	172.61	136.98	120.43
10	188.98	172.23	155.35	120.95	108.39
11	171.80	156.57	141.26	109.78	99.90
12	157.06	143.53	129.46	100.79	90.32
13	145.37	132.49	119.50	93.03	83.38
14	134.98	123.02	110.96	86.39	77.42
15	125.98	114.78	103.56	80.63	72.26
16	118.11	107.64	97.09	75.59	67.74
17	111.16	101.31	91.38	71.14	62.53
18	104.99	95.59	86.31	67.19	60.22
19	99.46	90.56	81.76	63.86	57.05
20	94.49	86.12	77.68	60.47	54.19
21	89.99	82.01	73.98	57.59	51.61
22	85.14	78.29	70.62	54.98	49.27
23	82.16	74.88	67.54	52.58	47.13
24	78.74	71.76	64.73	50.39	45.16
25	75.59	68.89	62.14	48.38	43.36
26	72.68	66.54	59.75	46.52	41.68
27	69.99	63.79	57.54	44.59	40.14
28	67.49	61.51	55.48	43.20	38.71
29	65.17	59.39	53.57	41.70	37.37
30	62.99	57.42	51.78	40.31	36.12
32	59.06	53.82	48.55	37.77	33.87
34	55.58	50.66	45.69	35.57	31.87
36	52.50	47.84	43.15	33.59	30.10
38	49.73	45.38	40.88	31.82	28.52
40	47.24	43.05	38.83	30.23	27.09
42	44.99	41.00	36.98	28.79	25.80

TABLE—(*Concluded*)

OF SAFE WORKING EXTERNAL PRESSURES ON FLUES 20 FEET LONG.

BIRMINGHAM GAUGE.	3	4	5	6	7	8
Thickness of Iron.	.259	.238	.220	.203	.180	.165
Diam. in In.						
6	150.25	126.87	108.40	92.30	72.57	60.98
7	128.78	108.75	92.92	79.11	62.20	52.27
8	112.68	95.15	81.30	69.22	54.43	45.73
9	100.16	84.58	70.42	61.53	48.38	40.65
10	90.15	76.12	65.04	55.38	43.54	36.58
11	81.95	69.20	59.13	50.35	39.58	33.25
12	75.12	63.44	54.20	46.15	36.28	30.48
13	69.35	58.56	50.03	42.60	33.49	28.14
14	64.39	54.37	46.46	39.55	31.10	26.13
15	60.10	50.75	43.36	36.91	29.02	24.39
16	56.34	47.58	40.65	34.61	27.21	23.05
17	53.03	44.78	38.25	32.57	25.61	21.52
18	50.08	42.29	36.13	30.76	24.18	20.32
19	47.45	40.07	34.23	29.14	22.91	19.25
20	45.08	38.06	32.52	27.68	21.71	18.29
21	42.93	36.24	30.96	26.37	20.73	17.42
22	40.98	34.60	29.56	25.17	19.79	16.62
23	39.19	33.09	28.27	24.07	18.93	15.90
24	37.56	31.71	27.10	23.07	18.14	15.24
25	36.05	30.44	26.05	22.15	17.41	14.63
26	34.67	29.27	25.01	21.29	16.74	14.45
27	33.38	28.19	24.08	20.51	16.12	13.55
28	32.19	27.18	23.22	19.77	15.55	13.06
29	31.08	26.24	22.42	19.09	15.01	12.61
30	30.04	25.37	21.68	18.45	14.51	12.19
32	28.17	23.78	20.32	17.30	13.60	11.43
34	26.51	22.38	19.12	16.28	12.80	10.76
36	25.04	21.19	18.06	15.38	12.09	10.16
38	23.72	20.03	17.11	14.57	11.45	9.62
40	22.53	19.03	16.26	13.84	10.88	9.14
42	21.06	18.12	15.48	13.18	10.36	8.71

Rule *for finding the Collapsing Pressure of Boiler Flues.*—Multiply the square of the thickness of the iron, in thirty-seconds of an inch, by the constant number 262·4; divide this product by the length of the flue in feet; divide this quotient by the diameter of the flue, in quarter feet, and the quotient will be the collapsing pressure in pounds per square inch.

EXAMPLE.

Diameter of flue, 24 inches.
Length of " 10 feet.
Thickness of iron, $\frac{3}{8}$ in.

Thickness, $\frac{3}{8} = \frac{12}{32}$.

Diam. 24 in. = 8 quarter ft.

$$
\begin{array}{r}
12 \\
12 \\
\hline
144 \\
262\,4 \\
\hline
576 \\
288 \\
864 \\
288 \\
\hline
10)37785{\cdot}6 \\
\hline
8)3778{\cdot}56 \\
\hline
472{\cdot}32 \text{ pounds.}
\end{array}
$$

Explanation of the following Tables of Collapsing Pressures.—The outside vertical column on the left-hand side of the table gives the length of the flue in feet; the horizontal column at the top of the table gives the diameter of the flue in inches. All the other columns denote the collapsing pressures in pounds per square inch.

COLLAPSING PRESSURE OF WROUGHT-IRON BOILER-FLUES $\frac{1}{4}$ INCH THICK.

Length in Feet.	Diameter of Flue in Inches.									
	9	12	15	18	21	24	27	30	33	36
4	1398	1049	840	699	600	525	466	420	382	349
6	932	698	560	466	400	349	311	285	254	233
8	699	524	420	350	300	262	233	210	191	175
10	559	419	336	280	240	210	186	168	152	140
12	466	349	285	233	200	175	155	142	127	116
14	400	300	239	200	171	150	133	120	109	100
16	350	262	210	175	150	131	117	105	95	87
18	311	233	187	156	133	116	103	95	85	78
20	278	210	168	140	120	105	93	84	76	70
22	255	190	153	128	109	95	85	76	69	64
24	233	174	142	116	100	87	78	71	63	58
26	215	160	129	107	82	80	72	64	58	54
28	200	150	120	100	85	75	67	60	54	50
30	186	140	112	93	80	70	62	56	51	46
32	175	131	105	87	75	65	58	52	47	44
34	164	123	99	82	70	61	55	49	45	41
36	155	116	94	77	66	58	51	47	42	39

COLLAPSING PRESSURE OF WROUGHT-IRON BOILER FLUES $\frac{5}{16}$ INCH THICK.

Length in Feet.	Diameter of Flue in Inches.									
	9	12	15	18	21	24	27	30	33	36
4	2187	1640	1312	1093	937	820	729	656	596	546
6	1458	1093	875	729	624	546	486	437	398	364
8	1093	820	656	546	468	410	364	328	298	273
10	875	656	525	437	374	328	292	262	238	218
12	729	546	437	364	312	273	243	218	199	182
14	625	468	375	312	267	324	208	182	170	156
16	546	410	328	273	234	205	182	164	149	136
18	486	364	292	243	208	182	162	146	133	121
20	437	328	262	218	187	164	146	131	119	109
22	398	298	238	199	170	149	133	119	108	99
24	364	273	218	182	156	136	121	109	99	91
26	336	252	202	168	144	126	112	101	92	84
28	312	234	187	156	133	117	104	93	85	78
30	291	218	175	145	125	109	97	87	79	72
32	273	205	164	136	117	102	91	82	74	68
34	257	193	154	128	110	96	86	77	70	64
36	243	182	146	121	104	91	81	73	66	60

COLLAPSING PRESSURE OF WROUGHT-IRON BOILER FLUES $\frac{3}{8}$ INCH THICK.

Length in Feet.	Diameter of Flue in Inches.									
	9	12	15	18	21	24	27	30	33	36
4	3148	2361	1889	1574	1349	1180	1049	944	858	787
6	2099	1574	1260	1049	900	787	699	630	572	525
8	1574	1180	944	787	674	590	525	472	429	393
10	1260	945	756	630	540	472	420	378	344	315
12	1050	787	630	520	450	393	350	315	286	260
14	900	674	540	450	385	337	300	270	245	225
16	787	590	472	393	337	295	262	236	214	196
18	699	525	420	350	300	262	233	210	191	175
20	630	472	378	315	270	236	215	189	172	157
22	573	429	343	286	245	214	191	171	156	143
24	525	393	315	262	225	196	175	157	143	131
26	484	363	292	242	207	181	161	146	132	121
28	450	337	270	225	192	168	150	135	122	112
30	420	315	252	210	180	157	140	126	115	105
32	393	295	236	196	168	147	131	118	107	98
34	370	278	222	185	158	139	123	111	101	92
36	350	262	210	175	150	131	117	105	95	87

COLLAPSING PRESSURE OF WROUGHT-IRON BOILER FLUES $\frac{7}{16}$ INCH THICK.

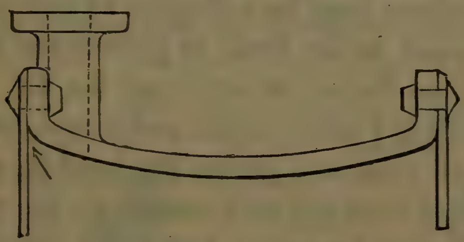
Length in Feet.	Diameter of Flue in Inches.									
	9	12	15	18	21	24	27	30	33	36
4	4786	3214	2571	2143	1837	1607	1428	1285	1169	1071
6	2857	2143	1714	1428	1224	1071	952	857	780	714
8	2143	1607	1285	1021	918	803	714	642	584	510
10	1714	1286	1028	857	735	643	571	514	468	428
12	1428	1071	857	714	612	535	476	428	390	357
14	1224	918	735	612	525	459	408	367	333	306
16	1071	803	642	536	459	401	357	321	292	268
18	952	714	571	476	408	357	317	285	260	238
20	857	643	514	428	367	321	289	257	234	214
22	779	584	468	390	334	292	260	234	212	195
24	714	536	428	357	306	268	238	214	195	178
26	659	494	396	330	282	247	220	198	180	165
28	612	459	367	306	262	230	204	183	167	153
30	571	428	342	285	245	214	190	171	156	142
32	536	401	321	268	229	201	179	160	146	134
34	504	378	302	252	216	189	168	151	137	126
36	476	357	285	238	204	178	159	142	130	119

BOILER-HEADS.

There are two forms of boiler-heads in general use, and four ways in which they are secured to the shell of the boiler. These are: 1st. The flat head turned outward. 2d. The flat head turned inward. 3d. The arched head turned outward. 4th. The arched head turned inward.

Considering the two facts, 1st, that, with a given amount of material, arched forms are stronger than flat ones, and, 2d, that cast-iron resists compressive better than tensile strains, it plainly appears that the first plan mentioned above is the weakest, and the fourth plan, the strongest way, a cast-iron head can be used. It is also true that either form of head is stronger when turned inward than otherwise.

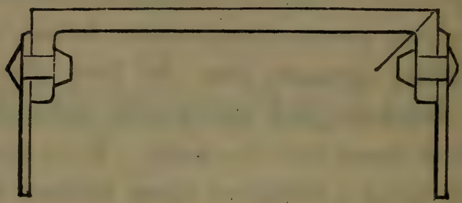
There is no doubt whatever as to the truthfulness of these statements in so far as the strength of the head is concerned, but there are other considerations besides strength which determine the form of boiler-heads.



Arched Head turned inwards.

The first to be considered is the arched head turned inward—the strongest plan. It will be noticed, that if the head is made of uniform thickness, with a curve at the spring-line of the arch, as it should be, to secure a sound casting, between the head and the sheet an acute angular space is left, liable to fill up with sediment and harden into scale by the action of the fire, which is usually severe at this part of the boiler.

Experience has shown that the boiler-plates at this point have corroded and burnt out very rapidly with the heads made and inserted in this manner, though the action of the sediment may be prevented by squaring up the head to a right angle with the sheet; but this renders the plate liable to over-heating from the excessive quantity of cast-iron in contact with it just over the fire.



Flat Head turned outwards.

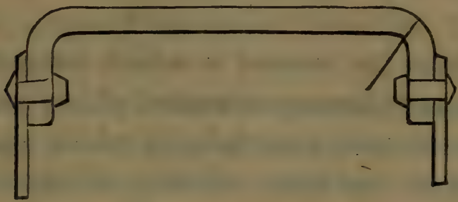
Now there are other objections to inserted heads, such as loss of capacity, and the necessity of tap-rivets over the water connections, which are more expensive to insert and more liable to leak than the usual form of rivets. Some of these objections may be overcome by setting the boiler far enough ahead in the front to protect the mass of iron in the head from the severe action of the fire.

Now, by adding $\frac{1}{4}$ more metal, and distributing it evenly in thickness all over, and giving the head an arched form, it can be turned outward, and possess all the requirements of strength needed for safety, and avoid the objectionable features of the concave head.

The flat head turned outward possesses the most objectionable features of any other form, as it is the worst disposition which can be made of the metal, to stand internal elastic pressure.

Whether boiler-heads be turned inward or outward, it is evident that they must possess strength equal at least to the metal of the sheet across the transverse rows of rivet-

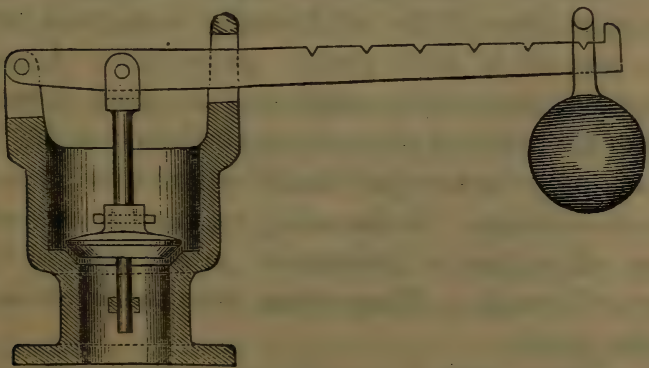
holes, as the section of metal after punching is the measure of strength in any boiler without stays.



Arched Head turned outwards.

While we can assume that the head loses the same amount of metal, by the rivet-holes, proportional to its thickness that the sheet does to which it is secured, whatever be the size or number of rivets, we have but to consider, in the comparison of strength, the ratio of thickness of head and sheet and the tensile strength of each material.

Wrought-iron heads of the flat, arched, and egg-shaped forms are now very generally used on account of their great tensile strength, lightness, and the facilities they afford to bracing, more particularly for boilers of large diameters.



SAFETY-VALVES.

The form and construction of this indispensable adjunct to the steam-boiler are of the highest importance, not only

for the preservation of life and property, which would, in the absence of that means of "safety," be constantly jeopardized, but also to secure the durability of the steam-boiler itself.

And yet, from the manner in which many things called safety-valves have been constructed of late years, it would appear that the true principle by which *safety* is sought to be secured by this most valuable adjunct, is either not well understood, or it is disregarded by many engineers and boiler-makers.

Boiler explosions have in many cases occurred when, to all appearance, the safety-valves attached have been in good working order; and juries, under the presidency of coroners, have not unfrequently been puzzled, and sometimes guided to erroneous verdicts, by scientific evidence adduced before them, tending to show that nothing was wrong with the safety-valves, and that the devastating catastrophies could not have resulted from over-pressure, because in such case the safety-valve would have prevented them.

It is supposed that a gradually increasing pressure can never take place if the safety-valve is in good working order, and if it have proper proportions. Upon this assumption, universally acquiesced in, when there is no accountable cause, explosions are attributed to the "sticking" of the valves, or to "bent" valve-stems, or inoperative valve-springs. As the safety-valve is the sole reliance in case of neglect or inattention on the part of the engineer or fireman, it is important to examine its mode of working closely.

The safety-valve is designed on the assumption that it will raise from its seat under the statical pressure in the boiler, when this pressure exceeds the exterior pressure on the valve, and that it will remain off its seat sufficiently

far to permit all the steam which the boiler can produce to escape around the edges of the valve. The problem then to be solved is, What amount of opening is necessary for the free escape of the steam from the boiler under a given pressure?

The area of a safety-valve is generally determined from ideas based on the velocity of the flow of steam under different pressures, or upon the results of experiments made to ascertain the area necessary for the flow of all the steam a boiler could produce under a given pressure. But as the fact is now generally recognized by engineers that valves do not rise appreciably from their seats under varying pressures, it is of importance that in practice the outlets around their edges should be greater than those based on theoretical considerations.

The next point to be considered is how high any safety-valve will rise under the influence of a given pressure. This question cannot be determined theoretically, but has been settled conclusively by Burg, of Vienna, who made careful experiments to determine the actual rise of safety-valves above their seats. His experiments show that the rise of the valve diminishes rapidly as the pressure increases.

TABLE

SHOWING THE RISE OF SAFETY VALVES, IN PARTS OF AN INCH, AT DIFFERENT PRESSURES.

Lbs. 12	Lbs. 20	Lbs. 35	Lbs. 45	Lbs. 50	Lbs. 60	Lbs. 70	Lbs. 80	Lbs. 90
$\frac{1}{36}$	$\frac{1}{48}$	$\frac{1}{54}$	$\frac{1}{65}$	$\frac{1}{86}$	$\frac{1}{86}$	$\frac{1}{132}$	$\frac{1}{168}$	$\frac{1}{168}$

Taking ordinary safety-valves, the average rise for pressures from 10 to 40 pounds is about $\frac{1}{40}$ of an inch, from

40 to 70 pounds about $\frac{1}{80}$, and from 70 to 90 pounds about $\frac{1}{120}$ of an inch.

The following table gives the result of a series of experiments made at the Novelty Iron-Works, New York, for the purpose of determining the exact area of opening necessary for safety-valves, for each square foot of heating surface, at different boiler pressures.

TABLE.

Pressure in Boiler in pounds above the Atmosphere.	Area of Orifice in square inch for each square foot of Heating Surface.	Pressure in Boiler in pounds above the Atmosphere.	Area of Orifice in square inch for each square foot of Heating Surface.	Pressure in Boiler in pounds above the Atmosphere.	Area of Orifice in square inch for each square foot of Heating Surface.
0.25	.022794	10	.005698	70	.001015
0.5	.021164	20	.003221	80	.000892
1	.018515	30	.002244	90	.000796
2	.014814	40	.001723	100	.000719
3	.012345	50	.001398	150	.000481
4	.010582	60	.001176	200	.000364
5	.009259				

The experiments were made on a tubular boiler with no means of escape for the steam except the safety-valve.

If we compare the area of openings, according to these experiments, with Zeuner's formula, which is entirely theoretical, it will be observed that the results from the two sources are almost identical, or so nearly so as not to make any very material difference.

In the absence of any generally recognized rule, it is customary for engineers and boiler-makers to proportion safety-valves according to the heating surface, grate sur-

TABLE

OF COMPARISON BETWEEN EXPERIMENTAL RESULTS AND THEORETICAL FORMULÆ.

Boiler Pressure, 45 Pounds.			Boiler Pressure, 75 Pounds.		
Heating Surface.	Area of Opening found by Experiment.	Area of Opening according to Formula.	Heating Surface.	Area of Opening found by Experiment.	Area of Opening according to Formula.
Sq. Ft.	Sq. Inches.	Sq. Inches.	Sq. Ft.	Sq. Inches.	Sq. Inches.
100	.089	.09	100	.12	.12
200	.180	.19	200	.24	.24
500	.45	.48	500	.59	.59
1000	.89	.94	1000	1.20	1.18
2000	1.78	1.90	2000	2.40	2.37
5000	4.46	4.75	5000	6.00	5.95

face, or horse-power of the boiler. While one allows 1 inch of area of safety-valve to 66 square feet of heating surface, another gives 1 inch area of safety-valve to every 4-horse power; while a third proportions his by the grate surface,—it being generally the custom in such cases to allow 1 inch area of safety-valve to $1\frac{3}{4}$ square feet of grate surface.

Experiments are very much needed to determine the proportions of safety-valves that would be capable of liberating all the steam which a boiler might produce with the fires in full blast, and all other means of escape closed. Until such a safety-valve shall be devised and adopted into general use, safety from gradually increasing pressure must depend to a certain extent on the watchfulness of engineers and firemen.

The safety-valve has not heretofore received that attention from engineers and inventors which its importance as a

means of safety deserves. In the construction of all other kinds of machinery, continual efforts have been made to insure accuracy; while in the case of the safety-valve, very little improvement has been made either in design or fitting. It is difficult to see why this should be so, when it is known that deviations from exactness, though small in themselves, when multiplied, become not only detrimental to the free action of the valve, but actually endanger the safety of the boiler.

Safety-valves should never be made with rigid stems, as, in consequence of the inaccuracy of other parts of the fitting, the rigid stem has a tendency to prevent the valve from adjusting itself on the seat, thereby causing leakage; as a remedy for which, through ignorance or want of skill, they are often jammed or overweighted, the stems being frequently bent, thus rendering the safety-valve a source of danger rather than a means of safety.

It is advisable, in all cases, to have the seating of safety-valves as narrow as the varying modes of construction will permit, as, with a wide seating, they are more liable to "stick," and more difficult to repair when leaky or out of order.

Safety-valve levers should, in all cases, be as short as the ordinary working pressure will permit.

Safety-valve levers should, in all cases, be moved before the fire is started under the boilers, in order to ascertain if the valves are in working order.

The safety-valve should never be raised when the water is dangerously low in the boiler.

All compound or complicated safety-valves should be avoided, as any interference with the direct action of the valve and lever has a tendency to render them unreliable.

The safety-valve should only be regarded as a means of safety when well proportioned, well constructed, and well cared for after being put in use.

The lift of the safety-valve, to give an opening equal to its area, should be one-half the radius or one-quarter the diameter.

RULES.

Rule *for finding the Weight necessary to put on a Safety-valve Lever, when the Area of Valve, Pressure, etc., are known.* — Multiply the area of valve by the pressure in pounds per square inch; multiply this product by the distance of the valve from the fulcrum; multiply the weight of the lever by one-half its length (or its centre of gravity); then multiply the weight of valve and stem by their distance from the fulcrum; add these last two products together, and subtract their sum from the first product, and divide the remainder by the length of the lever: the quotient will be the *weight* required.

EXAMPLE.

Area of valve, 12 inches.	65	13	8
Pressure, 65 pounds.	12	16	4
Fulcrum, 4 inches.	780	78	32
Length of lever, 32 inches.	4	13	
Weight of lever, 13 pounds.	3120	208	
Weight of valve and stem, 8 pounds.	240	32	
	32)2880	240	
	90 lbs.		

Rule *for finding the Pressure per Square Inch when the Area of Valve, Weight of Ball, etc., are known.* — Multiply the weight of ball by length of lever, and multiply the weight of lever by one-half its length (or its centre of gravity); then multiply the weight of valve and stem by their distance from the fulcrum. Add these three products

together. This sum, divided by the product of the area of valve, and its distance from the fulcrum, will give the pressure in pounds per square inch.

EXAMPLE.

Area of valve, 7 inches.	50	12	6
Fulcrum, 3 inches.	30	15	3
Length of lever, 30 inches.	1500	60	18
Weight of lever, 12 pounds.	180	12	
Weight of ball, 50 pounds.	18	180	
Weight of valve and stem, 6 pounds.			7
	21)1698		3
	80·85 lbs.		21

Rule *for finding the Pressure at which a Safety-valve is Weighted when the Length of Lever, Weight of Ball, etc., are known.*—Multiply the length of lever in inches by the weight of ball in pounds; then multiply the area of valve by its distance from the fulcrum; divide the former product by the latter: the quotient will be the pressure in pounds per square inch.

EXAMPLE.

Length of lever, 24 inches.	52	7
Weight of ball, 52 pounds.	24	3
Fulcrum, 3 inches.	208	21
Area of valve, 7 inches.	104	
	21)1248	
	59·42 lbs.	

The above rule, though very simple, cannot be said to be exactly correct, as it does not take into account the weight of the lever, valve, and stem.

Rule *for finding Centre of Gravity of Taper Levers for Safety-valves.*— Divide the length of lever by two (2); then divide the length of lever by six (6) and multiply the quotient by width of large end of lever less width of small end, divided by width of large end of lever plus width of small end. Subtract this product from the first quotient, and the remainder will be the distance in inches of the centre of gravity from large end of lever.

EXAMPLE.

Length of lever..36 inches.
Width of lever at large end.................................. 3 "
Width of lever at small end .... 2 "

$36 \div 2 = 18 - 1{\cdot}2 = 16{\cdot}8$ inch. $36 \div 6 = 6 \times 1 = 6 \div 5 = 1{\cdot}2$.

Centre of gravity from large end, 16·8 inches.

FOAMING.

The tendency of the water in a steam-boiler to rise into the cylinder is well known to engineers, and is generally attributed to the presence of dirt, grease, and other soapy substances; but, under certain circumstances, might be attributed to an insufficiency of steam room in the boiler, which would induce foaming, however clean the water might be.

Foaming is promoted, if not actually caused, by the reduction of pressure, and consequent ebullition of the water immediately below that point of the boiler whence the steam is drawn, which disposes the water, in the form of spray, to be carried along with the ascending current of steam. Not only is the water thus carried into the steam-pipe, but also any particles of earthy and other foreign matter that may happen to be at the broken surface of the water.

In analyzing the various causes concerned in the pro-

duction of foaming, it is necessary to take into consideration the effects produced by the necessarily intermittent action of the steam-valves. The supply of steam to the cylinder being cut off for a considerable period during each stroke, the effect is to throw the water in the boiler into a slight undulatory motion, as may frequently be observed in the glass water-gauge.

Foaming is also probably due in some measure to the flow of steam to the point of escape, carrying particles of water along with it by the induced current it produces.

The three most common causes of foaming are insufficient steam room, foulness of boilers, and excessive firing. Various expedients have been resorted to, such as perforated pipes, baffle-plates, etc., without any very beneficial effect; but experience has shown that the most practical and most reliable preventives of foaming are ample steam room, clean boilers, and moderate firing.

Foaming in locomotive boilers is generally caused by impurities in water, which are confined to certain parts of the country known as the alkali regions; these impurities are composed essentially of potash, soda, ammonia, and lithia. Locomotive boilers using surface water are also apt to foam if allowed to become dirty, in consequence of decayed vegetable matter held in suspension in the water, because such sedimentary accumulations add to the strength of the ingredients above referred to.

Foaming in marine boilers not unfrequently arises from insufficiency of steam room, as at each stroke of the engine a great portion of the steam is taken out, and as a result the pressure is lessened to such an extent as to induce violent foaming. Foaming is also inherent in some types of boilers, in consequence of their peculiar construction preventing a free escape of the steam from the heating surface to the steam room.

Foaming in marine boilers is most generally caused by changing the water from salt to fresh, or from fresh to salt, and is made evident by the boiling up of the water in the glass gauge. When foaming occurs from this cause, it is desirable to change the water in the boiler, and make it all of one kind, either salt or fresh, as soon as possible; and in order to do this, it is necessary to put on a strong feed and blow out slowly and continually, or at short intervals, until the water is changed.

It sometimes becomes necessary, when the foaming is very violent, to throttle down the steam, cut off short by means of the link, or even stop the engine, in order to ascertain the level of the water in the boilers, as it will always be found to be higher when the engine is in motion than when standing still.

It frequently occurs that, when the engine is stopped to suppress foaming, the water has fallen considerably below the proper level; under such circumstances, it is always better to dampen the fires, and start the independent feed-pump and pump in a fresh supply of water.

Boilers with a large amount of heating surface and small steam room generally foam; so also do boilers with the ordinary amount of steam room, if the water be carried too high.

All the phenomena connected with foaming have not yet been satisfactorily explained; but, from whatever cause it may arise, it is always attended with a certain amount of danger.

INCRUSTATION IN STEAM-BOILERS.

Most waters used for stationary and locomotive boilers contain solid matters in solution which become precipitated by elevation of temperature, or are left behind by evapora-

tion. On the matters ceasing to remain in solution, the first effect will be their deposition, and unless blown out sooner or later, the deposit becomes hardened and forms incrustation. The quantity of matters held in solution are commonly from 20 to 30 grains per gallon, and in some few cases reach as much as 100 grains per gallon.

The mere amount of solid matter in any water is no indication of its fitness, or otherwise, to be used in a steam-boiler, as this depends almost entirely on the nature of the solid impurities contained.

The presence of 50 grains per gallon of deliquescent salts—such, for example, as carbonate or chloride of soda—would not be seriously felt with a moderate amount of attention to blowing off; whereas, on the other hand, an equal quantity of salts of lime would render the water unfit for use, unless an unusual amount of care and attention were bestowed on blowing out and cleaning the boiler. Unfortunately, the presence of the former description of salts is the exception, whilst the latter is the rule.

It is generally understood that the carbonate of lime—the same substance, chemically speaking, as selenite, chalk, marble, and limestone—is held in solution in fresh water by an excess of carbonic acid, and that in reality it is present in the state of a bicarbonate. By heating the water, the excess of carbonic acid is driven off, and the greater part of the carbonate is precipitated.

Its solubility diminishes as the temperature increases, and at boiling-point it is scarcely soluble at all. It is for this reason that in water from which the air has been expelled, carbonate of lime is found in such small quantity. Carbonate of lime has been variously estimated as soluble in from 24,000 to 16,000 times its volume of water at ordinary temperature, or in the proportion of from $2\frac{3}{4}$ to $4\frac{1}{4}$ grains per gallon.

Sulphate of lime, a substance of the same chemical composition as gypsum, or plaster of Paris, is next in importance to carbonate of lime. Its solubility also varies greatly with the temperature; its greatest solubility is at 95° Fah., when it dissolves in 393 times the weight of water, or in the proportion of 178 grains to the gallon. At 212° it is only soluble in 460 times its weight of water, or 152 grains to the gallon; like carbonate of lime, it is completely insoluble at about 290°. It is, therefore, evident that these two salts are precipitated in all kinds of water merely by the elevation of temperature, when the boiler is worked at about 60 pounds pressure.

In boilers working at a low pressure, the sulphate of lime could be partially extracted by blowing off, if the water became saturated with it at about 230°; but its solution requires time, and the rapid evaporation precipitates it more rapidly than it can redissolve.

Carbonate of magnesia, or magnesian limestone, is the next important impurity in fresh water; but it usually exists in much smaller quantities than the other two salts. On its relation to temperature, and in its behavior in the water, it is similar to carbonate of lime.

On becoming insoluble, the lime and other salts remain for a time suspended in the water, and tend to deposit themselves more or less rapidly, according to the density of the water, the manner in which it circulates, and the intensity of the ebullition.

Over those parts of the heating surface where the water boils rapidly, the insoluble salts are held in suspension by the agitation until the ebullition subsides; or, when the circulation is good, they are carried away with the currents until a comparatively quiet part of the boiler is reached, when they are deposited on the plates and tubes.

The manner in which the precipitation comes about

is sometimes very remarkable, especially when the feed-water, at a high temperature, enters the boiler nearly at the point of saturation. In such cases the lime-salts are deposited as they pass through the apertures in the feed-pipe, and gather fast and thick on the adjacent plates.

It is by many supposed that the plates over the furnaces are most liable to become covered with a thick incrustation, as the greatest quantity of water is here evaporated. This is, however, seldom or never found to be the case, unless the circulation is very bad, as, for instance, over the flat, stayed crowns of locomotive fire-boxes.

In plain cylindrical and internally fired tubular boilers, the suspended matters in the water are driven off the plates by the ebullition, and carried to the part of the boiler where the circulation is most feeble, or the coolest part of the boiler.

When a considerable amount of incrustation is found over the fire in ordinary externally fired boilers, it is usually caused by the detached scale, which has fallen from the sides of the shell in pieces too heavy to be carried away by the circulation. The danger of overheating from this cause is one of the principal arguments against the practice of having a fierce heat under a boiler-shell, where the nature of the incrustation renders it liable to cover the furnace plates to any great degree.

The carrying away of the deposited matter by the ebullition and circulation is also retarded by the presence of grease or sticky matters in the water, which form a part with the impurities, that often prove too heavy or tenacious for removal by the currents in the boiler.

The sulphate of lime on depositing forms an amorphous crust more or less hard, according to the other ingredients in combination with it, and the heat to which it is exposed. The carbonate of lime and carbonate of

magnesia, on the other hand, usually deposit a loose, fine powder, forming a white sludge with the water.

When the deposited carbonate of lime is present in considerable quantity along with other impurities, it will remain soft for a length of time, and, if not exposed to too high a temperature when drying or emptying the boiler, will be converted into a floury powder of a light color. But if the boiler be blown out while the plates and brick-work in the flues are at a high temperature, the sludge often becomes baked hard; and it is to this circumstance that a great amount of the hard incrustation from both the sulphate and carbonate of lime is due.

When a boiler fed with water containing salts of lime is blown out cold, and the interior is examined before it becomes dry, the plates, tubes, and stays may be found covered with a thick coating of light-colored slushy matter, that can be removed with very little trouble if brushed off or washed out with a hose-pipe and jet of water. Should, however, the interior be maintained at a high temperature by blowing out before the boiler and flues are cool, the deposit becomes baked on, and apparently there is not so much left for removal.

Various attempts have been made to calculate the loss of heat caused by incrustation formed on the heating surface. But the circumstances to be considered which determine the rate of heat transmission through plates covered by scale of different kinds and thickness, either homogeneous or otherwise, are not sufficiently well understood, and are too numerous to admit of anything like exact calculation. But it has been very satisfactorily proved by observation and experiment that $\frac{1}{16}$ inch of incrustation on the tubes of a boiler is equivalent to a loss of 20 per cent. of fuel, and that the loss increases in a very rapid ratio.

It frequently happens that no two of the numerous layers are alike in color, consistency, or chemical composition,—a fact due to the disturbing influence at the source of the feed supply. The face of the incrustation next to the plate is very often of a black color, and adhering to it is found a film of oxide of iron, whilst the surface of the plate is quite soft, and bears unmistakable signs of wasting, sometimes to a considerable depth. This is usually caused by the corrosive action of the iron salts, and in brackish water by chloride of magnesia (muriate of magnesium). This last salt is the destructive agent in sea-water.

From water containing salts of iron in considerable quantity the incrustation formed has often a red tinge. Chalybeate waters are generally highly injurious to the plates, and the film of incrustation next to the iron is sometimes of a deep red, coloring the water that comes in contact with it through the fissures in the scale, by which the presence of these injurious salts of iron is easily detected.

When the water in a steam-boiler becomes impregnated with the above-named ingredients, great resistance is offered to the free escape of the steam bubbles and to the free convection of heat. The water is, in consequence, lifted off the plates by the steam that accumulates on their surface, and allows them to become over-heated.

The tendency to over-heating is much aggravated, if grease or other organic matter be present in the water along with this floury deposit. The grease appears to combine mechanically with the carbonate of lime, and when the compound sinks on to the plates over night, or when the boiler is at rest, it clings as a loose, spongy mass, too inert to be carried off by the circulation or ebullition which it retards, and by preventing the contact between the plates and the water, and by offering great resistance

to the transmission of heat, produces over-heating of the plates.

There are but few problems connected with steam engineering at which inventors have tried their hands to a greater extent, than the prevention and removal of boiler incrustations. Numerous inventions, and not less than three hundred patents, have been taken out for that purpose, but most of them have proved futile,—some because their inventors did not fully comprehend the magnitude of the object involved, and others on account of the diversity of conditions and circumstances under which they have been tried.

The substances used to act mechanically in preventing and removing incrustation by decreasing the cohesion and adhesion of the deposited particles, are even more numerous than those employed to act chemically in decomposing and dissolving the solid matters. In fact, it is difficult to mention any common commodity that has not been employed to prevent incrustation in one way or the other, although the manner in which different substances may act is often not understood by those who employ them.

The following chemical and mechanical remedies have been proposed at different times for the removal and prevention of scale in steam-boilers:

Frequent blowing off.

Employment of some collecting apparatus.

Thorough circulation of the water in the boiler.

Purification of the water before entering the boiler.

Cracking off the scale by expansion.

Employment of galvanic batteries or electric anti-crustators.

As before stated, what is needed to render efficient and permanent relief is an article that will attack the scale, render it porous, and destroy the affinity between it and

the iron, without any injuries to the latter, and will hold the minerals and ingredients, which are passing in with the feed-water, in the form of slush or sludge, until they can be blown out. G.W. Lord, a practical manufacturing chemist of Philadelphia, who has been, at various times, connected with many mechanical enterprises in this country, the West Indies, and South America, has succeeded, by experiment and observation, in producing an article—Lord's patent boiler compound — which has been in use over eight years in all parts of the United States, Canada, South America, Mexico, and Cuba, under the most varying circumstances, and in all cases with satisfactory results. The manufacturer and patentee can produce more than ten thousand testimonials of its efficiency from engineers and steam-users. It neutralizes mine and mineral waters, which contain lime, iron, sulphur, and carbonates, destroys their affinity, and renders them simple and harmless. It not only prevents the formation of new scale, but decomposes the old and converts it into a soluble sediment, which may be blown out every day. It contains no acid which has any injurious effect on the iron of the boiler, — evidence of which may be found in the fact that the manufacturer, some years ago, filled several thousand vials with a solution of his compound, in which was placed a quantity of bright iron turnings and small pieces of steel wire, which appear as bright as the day they were immersed in the solution, one of which will be sent to any one who feels incredulous on the subject. Lord's compound gives relief in all cases when used according to directions. Parties wishing to test its efficiency should address Geo. W. Lord, Philadelphia, Pa.

INTERNAL AND EXTERNAL CORROSION OF STEAM-BOILERS.

Internal and external corrosion are the two maladies that boilers are most liable to suffer from.

Internal corrosion presents itself in various forms, each having a peculiar character of its own, though only sometimes strongly marked; these are designated as *uniform corrosion, wasting, pitting, honey-combing,* and *grooving.*

External corrosion is said to be due to galvanic action, or the influence of chemicals and dampness combined.

By uniform corrosion is meant that description of wasting of the plates or tubes, where the water corrodes them, in a more or less uniform manner, in patches of considerable extent, and where there is usually no well-defined line between the corroded part and the sound plate.

The presence of this as well as other kinds of corrosion can generally be easily detected, even when covered with a considerable thickness of incrustation, as its presence is often revealed on emptying the boiler by the bleeding, or red streaks, where the scale is cracked; although in some cases, even where the plates are free from incrustation, uniform corrosion, in consequence of its even surface and the absence of any well-defined limit to its extent, may sometimes escape detection.

Even when actually discovered, the depth to which it has penetrated can only be ascertained by drilling holes through the plate and measuring the amount of material remaining. With lap-joints, the thickness remaining at the edge of the plate and round the rivet-heads may serve as a guide to the amount of wasting; but this may prove treacherous, since the adjacent plates may both be corroded to an equal extent along with the rivet-heads, which

will give the edge of the plate the appearance of having the original thickness.

Another peculiarity worthy of notice is the different manner in which the plates and rivet-heads are affected by different kinds of waters after the wasting has been going on for some time. In most cases the corroded iron is readily removed, if it does not come off without means being taken to detach it. But cases are to be met with where the corroded iron adheres tenaciously to the sound plate beneath. In such cases considerable force is required to remove it, and the presence of the corrosion is not suspected until the hammer or pick is forcibly applied.

It frequently occurs that in the case of two boilers alike in every respect, fed with the same water, and subject to the same treatment, one may be found attacked at the front end, whilst the other may be affected only on the bottom at the back end.

With the feed-water from one supply only, corrosion is found more often under an incrustation of sulphate of lime than under one consisting chiefly of carbonate of lime. In many boilers fed with water containing the former salt, a coating of oxide of iron of a black color may be found adhering to the detached scale, which, as often as it reforms and is broken off, brings with it a fresh film of oxide.

Various means, such as the use of rain, surface, and distilled waters, have been employed for the prevention of internal corrosion, but were found impracticable and generally abandoned, as the expense involved in most cases was found to exceed that of replacing the corroded boiler with a new one, even after a service of only a few years.

Lord's Boiler Compound appears to be the only known remedy that affords any protection to boilers against the

fearful effects of this singular and mysterious phenomena, as it has been found to neutralize the mineral ingredients of the most destructive waters, and prevent the internal corrosion and wasting of boiler-plates, seams, and rivets.

External corrosion is frequently more destructive than internal, particularly in the case of stationary boilers. This probably arises from the fact that its presence is less suspected, and is often less easily detected in consequence of the covering of brickwork or other material surrounding the shell.

The most frequent sources of external corrosion are, exposure to the weather, leakage from seams, dripping from safety or other valves, moisture rising from the ground, either from the damp nature of the location or from the want of proper appliances to carry off the waste water.

A slight leakage from a bad joint may be sufficient to cause a severe local grooving at the seam or flange, as it often goes on for a length of time unperceived and unsuspected, especially when the shell is covered by brickwork, or other material to prevent the radiation of heat, as in such cases, if a leak takes place on the upper side of the boiler, the whole circumference of the shell is liable to suffer from it.

One of the most remarkable circumstances connected with all kinds of corrosion is the singular manner in which they make their appearance and act, affecting very few boilers alike, or even in the same locality.

Corrosion of Marine Boilers.—Marine boilers seldom last more than four or five years; whereas land boilers, made of the same quality of iron, often last fifteen or twenty years; yet the difference in durability is not the effect of any chemical action upon the iron by the contact of sea-water, for the flues of marine boilers rarely show

any deterioration from this cause; and even in worn-out marine boilers, the hammer-marks on the flues are as conspicuous as at the time of their formation.

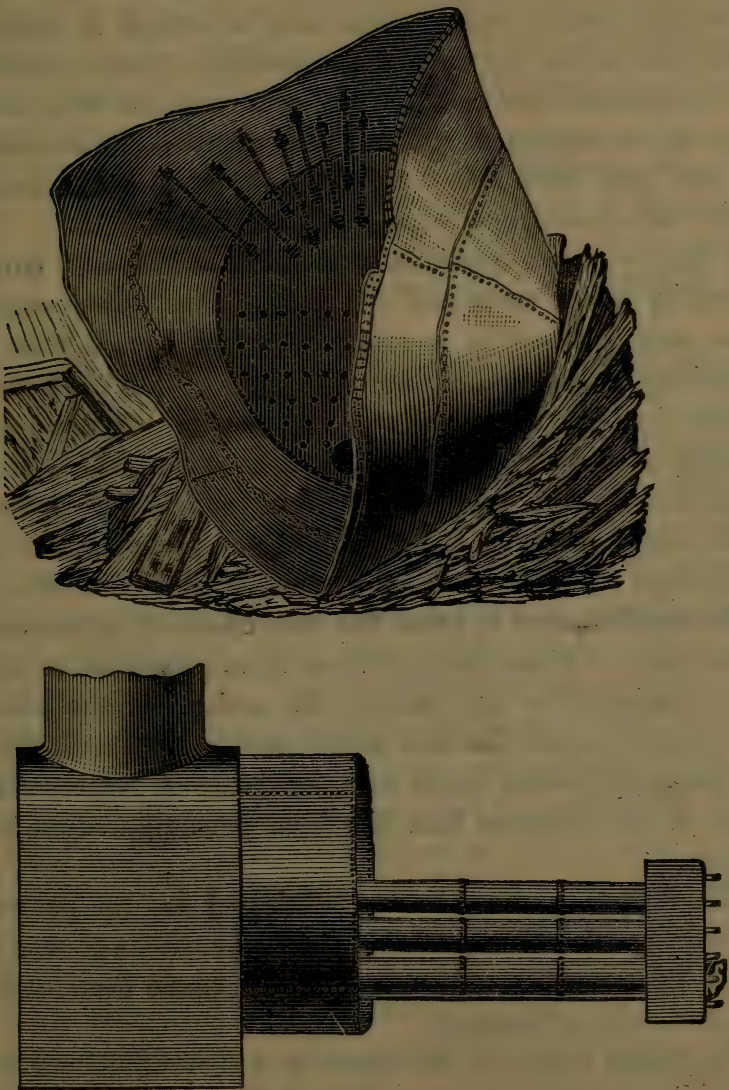
The thin film or scale spread over the internal parts of marine boilers would seem of itself to preserve that part of the iron from corrosion which is situated below the water-level; but, strange as it may seem, it is rare to find any internal corrosion of boilers using salt-water, in those parts of the boiler with which the water comes in contact; the cause, therefore, of the rapid corroding of marine boilers is not traceable to the chemical action of salt-water, as steamships provided with surface condensers, which supply the boilers with fresh water, have not reaped much benefit in the durability of their boilers.

Corrosion of steam-boilers is one of the most obscure subjects in the whole range of engineering.

BOILER EXPLOSIONS.

That the use of steam-power is fraught with danger is only too well known; the extent of the danger, however, as indicated by the number of boiler explosions every year, and the loss of life and property entailed, is but vaguely appreciated by the public at large. No official record is kept of such accidents, and only those of exceptional interest are reported in the newspapers. Even in such cases as are reported, it is almost impossible to ascertain their true cause, as there is seldom a unanimous opinion on the part of the experts who examine into the causes after the event.

There are a great many people who think they know something that will explain the cause of these fearful accidents, but, for some reason or other, their fine-spun theories have been of no practical value. One reason



EXPLODED BOILER OF THE FERRY-BOAT "WESTFIELD."

for this is, doubtless, that the conditions under which experiments have been made to determine the causes of explosions are entirely different from those under which boilers are used. Some points may be settled by experiment, such as the strength of the material of which boilers are constructed; but even here there is room for error, inasmuch as the conditions under which experiments are made on iron and steel are very different from those under which boilers are torn to pieces by explosion.

Formerly, the impression was almost universally entertained by steam users and the public that boiler explosions were induced by certain mysterious causes, such as electricity, generation of explosive gases within the boiler, caused by the decomposition of steam; instantaneous flashing of large bodies of water into steam, which were attempted to be explained by the spheroidal theory; great deterioration in the quality of the plates, caused by mysterious chemical changes, concussive ebullition, etc.

An unwillingness to know the true cause of explosions, on the part of steam users or those most interested, as well as an inability on the part of the authors of the above enumerated mysterious and occult causes to explain them satisfactorily, has no doubt been the means of perpetuating much of the nonsense that has been promulgated on this subject.

The following remarks are submitted for the purpose of showing the fallacy of the above theories, and also to direct the attention of engineers and steam users to the actual sources of danger.

Electricity might be developed in a steam-boiler under certain conditions; but it is difficult to conceive how any quantity could be accumulated, as Faraday, the eminent chemist, has proved that the development of electricity in a vessel containing steam was due solely to the friction of

the steam against the sides of the vessel. Such being the case, the presence of electricty would be most likely to occur in the pipe between the boiler and the engine, and would be induced by the friction of the escaping current; but even in that case, the steam must be wet, as the same authority proved that electricity could not be developed from a current of dry steam.

Admitting that the presence of electricity in an ordinary boiler is not impossible, it yet remains to be shown that it could exist in a state of high tension; and yet, again, how it could bring about an explosion, accompanied by the usual well known results.

Concussive Ebullition.—The phenomena called concussive ebullition, arises, according to Dufour, from the principle, that in order that a liquid may be transformed to vapor at any temperature, some portion of the surface must be freely exposed to a space into which the vapor may expand. This was demonstrated by suspending drops of water in heated oil. The temperature of the water was raised considerably above the boiling-point without the formation of vapor; but if a bubble of air or a piece of porous substance was placed in contact with the water, a burst of vapor occurred. But experiments with drops of oil or drops of water can, at best, shed but very feeble light on the causes of boiler explosions, as experiments in the laboratory are made under very different conditions from those to which steam-boilers are daily subjected in mills and factories.

Generation of Explosive Gases. —That a small quantity of steam might be decomposed in a boiler, by coming in contact with plates that have accidentally become red-hot, cannot be disputed; but that the decomposition could occur to any considerable extent with oxidized plates is wellnigh impossible. The hydrogen liberated by the de-

composition is not explosive, and would require to be united and intimately mixed with its equivalent of oxygen, and then ignited to produce an explosion.

Supposing the oxygen to be admitted with the feed-water, and that the ignition could be effected by red-hot plates or an electric spark, it still remains to be shown how the gases could possibly become so intimately mixed, in the presence of so large a body of steam and nitrogen present in the boiler, as to form a detonating compound.

Again, assuming that nearly all the steam could be decomposed, the hydrogen would only burn quietly in the presence of oxygen as it becomes liberated on the red-hot surface of the plates; and in any case, its power to produce an explosion is extremely improbable.

But to take the most extreme view of the case, and assuming the sudden formation of a vacuum within the boiler, by the union of the two gases, to take place, it is still by no means clear how the bursting of the shell would follow in consequence, as the vacuum formed could only be local and insignificant with a large quantity of steam and nitrogen in the boiler.

Spheroidal Theory.—The spheroidal theory is the so-claimed tendency of water, when thrown upon highly-heated plates, to assume the spheroidal condition and to evaporate suddenly when the temperature is sufficiently lowered. But the exact application of this theory is, however, by no means clear; and the assumed delay of the water in evaporating is antagonistic to the sudden evaporation from the overheating theory, as it is difficult to see how the evaporation of a large quantity of water in an ordinary boiler could be delayed, as is assumed in this theory, without reducing the temperature of the water below that sufficient to produce an explosion.

Deterioration of Plates.—This subject admits of no

speculation, as it comes under the head of "wear and tear," which has been discussed in a former article. That the sudden heating or cooling and oxidation of parts of the boiler induce a great deterioration of strength has been proved by experience, but such evils can only be avoided by constant attention and repairs.

Explosions can occur from one cause only — deficiency of strength in the shell or other parts of a boiler. This deficiency of strength may be an original defect arising in the material or workmanship at the time of construction, or, it may be due to deterioration from use, from ordinary wear, or from injuries occurring from mismanagement, want of attention, and repairs, etc.

It often happens that boilers are deficient in strength for the pressure they are intended to bear, and no accumulation of pressure beyond this is necessary to bring about their destruction; the circumstance of a boiler being unable to stand its ordinary working pressure may be due in part to its original design, and also to the ignorance of those who fixed the pressure it was worked at, its design and power of resistance not being fully understood.

Defects may arise from workmanship or material, whose presence in a great majority of cases can be detected by proper inspection and testing, but which may, in some cases, escape the closest scrutiny, furnishing an additional evidence of the degree of watchfulness necessary on the part of engineers and those having the care and management of steam-boilers.

The defects of workmanship are most liable to escape detection in tubular boilers and in those of the locomotive type, where the inside cannot be examined unless the tubes are removed, or the boiler partly taken to pieces.

Defects of material, such as blisters, lamination, and the adhesion of sand or cinders in rolling, can sometimes,

but not always, be detected by inspection. Brittleness of material, unless it be glaringly bad, can seldom be discovered by ordinary inspection after the construction of the boiler is completed.

Over-pressure.—Boilers are not unfrequently found running by the steam-gauge at a certain pressure which is regarded perfectly safe; but when the gauge is examined and compared with one known to be correct, it is found to be 10, 20, or perhaps, as is sometimes the case, 50 pounds out of the way. If a boiler supposed to be running under a pressure of 80 pounds is found, in consequence of an unreliable steam-gauge, to be actually running at a pressure of 120 to 130 pounds, the limit of safety may have been passed, and an accident is imminent, which may occur at any moment.

Over-pressure may also be caused by the safety-valve being recklessly overweighted, by the sticking of the valve on its seat, or by the inadequate size of the communication between the boiler and the safety-valve, and also by placing the safety-valve on a branch-pipe between different boilers.

Overheating.—There is no doubt that exposure of the upper surfaces of flues or the crown of a furnace to the intense action of heat, when there is no water upon their surfaces to absorb or transfer this heat, is highly injurious and destructive to the boiler; and on this ground alone all the devices for regulating or observing the water-level are necessary and advisable.

A boiler may be well-designed and made up of good material and first-class workmanship, and yet in a few months after being put under steam, it may explode with terrible effect. On examining into the cause of the explosion, it may turn out that the water which was used, made a heavy deposit; that the boiler had not been

cleaned out since it was put in use; that the fires had been fiercely urged and the water driven from the surface of the iron; as a result, the life had been entirely burnt out of the sheets directly over and around the fire, thereby weakening the boiler and putting it in a dangerous condition.

Overheating may also be due to shortness of water, which in turn may be induced by leakage of valves, stop-cocks, mud- or hand-holes below the water-line; or by excessive priming in boilers containing little water; or it may be the result of failure in the feed-pipe supply, or neglect to start the water at the proper time, or turn it on in sufficient quantity.

Shortness of water may result from the check-valve being kept from its seat by dirt, shavings, or straw, lifted by the pump from the well or cistern; under such circumstances the pressure in the boiler would force the water back under the valve into the pump-barrel, from which it would escape if the drip- or pet-cocks were carelessly or inadvertently left open.

Accumulation of Deposits.—Explosions in many cases are the result of the accumulation of deposits in boilers, as the deposit not unfrequently takes the form of hard, solid incrustation, which prevents the water from absorbing or neutralizing the effect of the heat transmitted from the fire to the boiler.

The presence of such deposits is generally manifested by leakage at the seams, and bulging of the plates directly over the fire, when the boiler may be considered permanently injured, and liable to explode at any time.

Excessive Firing.—Excessive firing is also the cause of many disastrous explosions, and occurs most frequently where the boiler is too small for the engine, or incapable of furnishing the required amount of steam, as the intensity

of the fire necessary to generate the desired quantity of steam has a tendency to repel the water from the plates. The same effect may be produced when there is a great disproportion between the grate and heating surfaces, or where the heat from a large grate is concentrated on a small space. Under such circumstances, the heat is delivered with such intensity as to lift the water from the surface of the iron, thereby exposing it to the direct action of the fire. Explosions occurring from excessive firing are in all cases the result of avarice, ignorance, or a want of skill in the care and management of the steam-boiler.

It has been shown, in the examination of this subject, that no amount of theory will prevent boiler explosions, nor will any number of experiments in the laboratory or on obsolete types of boilers be of any use whatever in deciding what was the cause of such an accident, as all the conditions under which the boiler exploded must be considered,— the material of which they are constructed, workmanship in construction, form or type of boiler, setting attachments, quality of water used, kind of fuel, and last, but by no means least, skill employed in their care and management. These are the vital points, and the ones to be considered in order to arrive at any approximate solution of the cause or causes of steam-boiler explosions.

Mainly through the operations and researches of the Hartford Steam-Boiler Inspection and Insurance Company, boiler explosions have been stripped of the mystery in which they were to a certain extent enshrouded, and ascribed to their true causes; and in view of the numerous defects that tend directly and indirectly to produce explosions, that are almost daily brought to light by the trained inspectors of that Company, the mystery to be solved, if there be any mystery connected with boiler exsions, seems to be why more boilers do not explode, even at their ordinary working pressures.

The fidelity and skill with which the inspections are made by that Company, as well as the correctness of the theory on which they are based,—a theory which discards mysteries in accounting for boiler explosions,—are sufficiently attested by the almost entire absence of serious accidents in connection with the thousands of boilers of all sorts and conditions that are or have been in their care.

Whenever the yearly reports of the Hartford Steam-Boiler Inspection and Insurance Company are collected, and given to steam users and the public in book-form, they will form one of the finest contributions to the scientific literature of the country that has ever been heretofore published, as they will contain an immense amount of important and practicable information on the subject of steam-boilers and steam-boiler explosions which could be obtained from no other source.

There is no mystery about steam-boiler explosions; they are all cause and effect; and it will be found, on investigation, that seven-tenths of all the boiler explosions that occur yearly in this country might be traced to some sufficient cause, were all the facts known. And every attempt to ascribe boiler explosions to obscure and mysterious causes can only be productive of mischief, as they engender carelessness on the part of owners and attendants, who are often led to believe that no amount of care on their part will avail against certain mysterious agents at work within their boilers.

COMPARATIVE STRENGTH OF SINGLE- AND DOUBLE-RIVETED SEAMS.

On comparing the strength of plates with their riveted joints, it will be necessary to examine the sectional areas,

taken in a line through the rivet-holes with the section of the plates themselves.

It is perfectly obvious that in perforating a line of holes along the edge of a plate, we must reduce the strength; it is also clear that the plate so perforated will be to the plate itself nearly as the areas of their respective sections, with a small deduction for the irregularities of the pressure of the rivets upon the plate; or, in other words, the joint will be reduced in strength somewhat more than in the ratio of its section through that line to the solid section of the plate.

It is also evident that the rivets cannot add to the strength of the plates, their object being to keep the two surfaces of the lap in contact.

When this great deterioration of strength at the joint is taken into account, it cannot but be of the greatest importance that in structures subject to such violent strains as boilers, the strongest method of riveting should be adopted. To ascertain this, a long series of experiments were undertaken by Mr. Fairbairn.

There are two kinds of lap-joints—single and double riveted, as shown in Figs. 1 and 2 on opposite page. In the early days of steam-boiler construction, the former were almost universally employed; but the greater strength of the latter has since led to their general adoption for all boilers intended to sustain a high steam pressure.

A riveted joint generally gives way either by shearing off the rivets in the middle of their length, or by tearing through one of the plates in the line of the rivets.

In a perfect joint, the rivets should be on the point of shearing just as the plates were about to tear; but in practice, the rivets are usually made slightly too strong. Hence, it is an established rule to employ a certain number of rivets per lineal foot, which, for ordinary diameters

and average thickness of plate, are about 6 per foot or 2 inches from centre to centre; for larger diameters and heavier iron the distance between the centres is generally increased to, say $2\frac{1}{8}$ or $2\frac{1}{4}$ inches; but in such cases it is also necessary to increase the diameter of the rivet, for while $\frac{5}{8}$ rivets, or even $\frac{1}{2}$ inch, will answer for small diameters and light plate, with large diameters and heavy plate experience has shown it to be necessary to use $\frac{3}{4}$ to $\frac{7}{8}$ rivets.

If these are placed in a single row, the rivet-holes so nearly approach each other that the strength of the plates is much reduced; but if they are arranged in two lines, a greater number may be used, and yet more space left between the holes, and greater strength and stiffness imparted to the plates at the joint.

Taking the value of the plate, before being punched, at 100, by punching the plate loses 44 per cent. of its strength; and, as a result, single-riveted seams are equal to 56 per cent., and double-riveted seams to 70 per cent. of the original strength of the plate.

It has been shown by very extensive experiments at the Brooklyn Navy Yard, and also at the Stevens Institute of Technology, Hoboken, N. J., that double-riveted seams are from 16 to 20 per cent. stronger than single-riveted seams — the material and workmanship being the same in both cases.

Fig. 1.

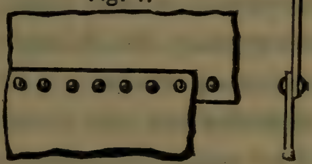
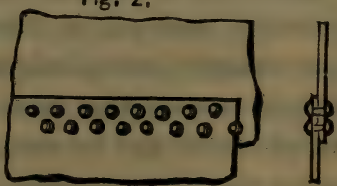


Fig. 2.



Taking the strength of the plate at........................ 100
The strength of the double-riveted joint would then be 70
And the strength of the single-riveted joint would be 56

CALKING.

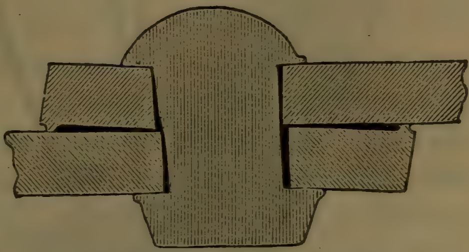
The object of calking is to bring together the seams of a boiler, after riveting, in such a manner that they shall be perfectly steam- and water-tight. This is done by using a sharp tool ground to a slight angle. The edge of the plates being first chipped or planed to an angle of about 110°, the calking-tool is then applied to the lower edge of the chipped or planed angle in order to drive or upset the edge, thus bringing the plates together, and rendering the joint to all appearances perfectly steam-tight and able to resist the internal pressure brought to bear upon this particular point.

It is well known that the use of a hammer on wrought-iron will granulate, or harden, it to such an extent as to make it almost as hard as steel. Now the angled tool before mentioned, through its action, in the process of calking, upon the lower edge of the chipped plate, causes a granulation of that plate; while the under one is much softer, in consequence of not being exposed to the action of the tool, consequently, the skin or outer surface of the softer material is indented or cut.

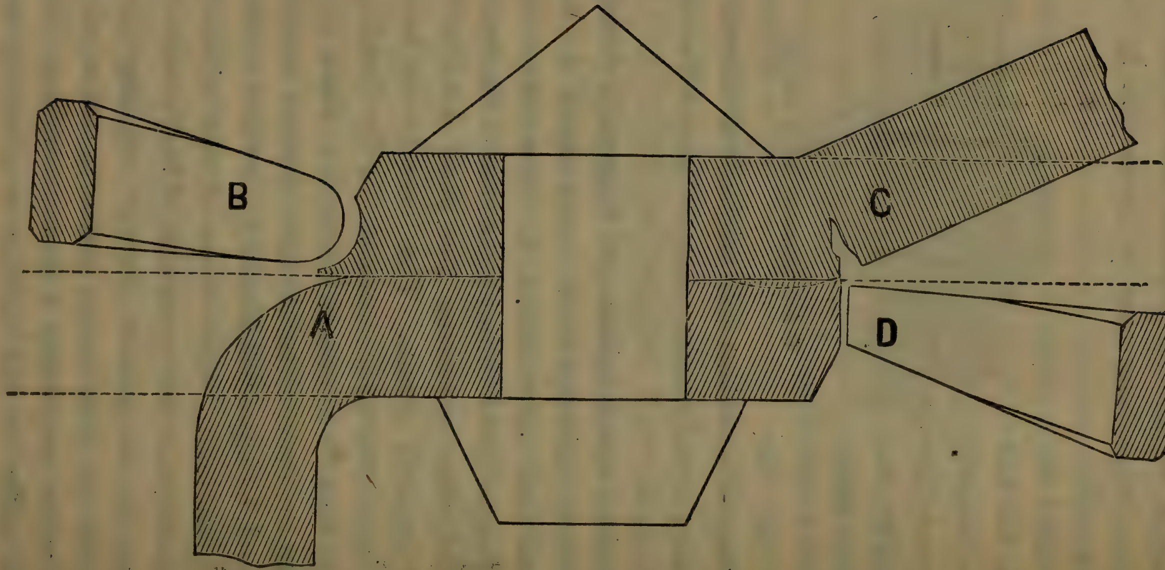
A boiler may be constructed by parties of high repute, and be made of the best material, and to all appearance be capable of standing any test that can be applied to prove its safety, and yet its durability may be very limited, or it may collapse or explode soon after being put in use, for the simple reason that a cause existed from the very first which could not be seen, nor any test point out, and that cause was the grooving or indentation made by the calking, which became larger and larger through corrosion, expansion, and contraction, thus rendering the plates unfit to resist the strain, which must eventually induce

rupture or explosion, resulting in loss of life and destruction of property.

This tendency to weaken the plates of steam-boilers by the present mode of calking may be illustrated by very familiar examples. When a blacksmith desires to break his bar of iron to a given length, he first cuts around the bar, weakening it; the breaking is then easily accomplished. A glazier similarly uses his diamond. These illustrations are perfectly analogous to that of the cutting or indentation made by the old-fashioned calking-tool.


Old-fashioned method of calking.

On examination, steam-boilers are frequently found to be fractured along the edge of the outer lap of the sheet both transverse and longitudinal, in consequence of a channel being entirely cut through the skin of the iron by the calking-tool, thus rendering the plate weak at the point of the greatest strain. The force to act is ever present; the iron is already strained, for by bending a sheet of iron to make a required circle, the fibres of the iron composing the outer circumference must, of necessity, be stretched, and, by imperfect bending, stretched laterally as well as longitudinally, while those of that composing the inner circumference are upset, and, if badly welded in the act of manufacture, pucker, thereby exposing the inside particles of the iron to the corrosive action of the acids in the water, producing a honey-combing. Thus everything is ready for the cutting or grooving to be made; the strain on the outer, the puckering on the inner,



CONNERY'S CONCAVE METHOD OF CALKING.

The annexed cut represents Connery's concave method of calking, described on page 467. B shows the concave tool; A, the plate bent after being calked by this method; D, the old-fashioned calking-tool; C, the plate bent after being calked by the old method.

circumference; it then only becomes a mere question of time as to the result.

Very few, except those familiar with the laws of steam, have any idea of the immense pressure exerted on the shells of steam-boilers under pressure;* and when we consider that this immense pressure is brought to bear along the lap of the joints,— the points deviating farthest from the true cylindrical form,— the importance of having the iron not only of good quality, but free from the defects induced by inferior calking, must at once be admitted. Immense sums of money have been expended in experiments with the object of ascertaining, if possible, the cause of boiler explosions, which, if conducted by competent persons, might have proved in many instances to be the result of a mischievous system of calking.

The cut on opposite page represents an improved method of calking, which is acknowledged by competent parties to be one of the most important improvements ever heretofore made in the construction of steam-boilers. It is the invention of James W. Connery, foreman of the Boiler Department at the Baldwin Locomotive Works, Phila., and is known as Connery's Concave Calking. By this method the dangers to life and property induced by the old system of calking are entirely obviated, as even the uninitiated cannot dent or gall the plates with Connery's Patent Calking; the importance of which will be appreciated by all steam users, more especially when it is known that it is impossible, for even the most skilful boiler-maker, to calk a boiler with the old-fashioned calking-tools without permanent injury to the plates. An illustration of which may be seen on page 465. The process of calking is also so simplified by Connery's Concave Method as to give it, in point of economy, claims to universal adoption.

* See page 389.

STRENGTH OF THE STAYED AND FLAT SURFACES.

The sheets that form the sides of fire-boxes are necessarily exposed to a vast pressure, and consequently some expedient has to be devised to prevent the metal at these parts from bulging out.

Stay-bolts are generally placed at a distance of 4½ inches from centre to centre all over the surface of fire-boxes, and thus the expansion or bulging of one side is prevented by the stiffness or rigidity of the other.

Now, in an arrangement of this kind, it becomes necessary to pay considerable attention to the tensile strength of the stay-bolts employed for the above purpose, since the question of the ultimate strength of this part of the boiler is now transferred to them, it being impossible that the boiler-plates should give way unless the stay-bolts break in the first instance.

Accordingly, all the experiments that have been made, by way of test, of the strength of stay-bolts, possess the greatest interest for the practical engineer. Mr. Fairbairn's experiments are particularly valuable. He constructed two flat boxes, 22 inches square. The top and bottom plates of one were formed of ½ inch *copper*, and of the other $\frac{3}{8}$ inch *iron*. There was a 2½ inch water-space to each, with $\frac{13}{16}$ inch iron stays screwed into the plates and riveted on the ends. In the first box, the stays were placed five inches from centre to centre, and the two boxes tested by hydraulic pressure.

In the copper box, the sides commenced to bulge at 450 pounds pressure to the square inch; and at 810 pounds pressure to the square inch the box burst, by drawing the head of one of the stays through the copper plate.

In the second box, the stays were placed at 4 inch centres; the bulging commenced at 515 pounds pressure

to the square inch. The pressure was continually augmented up to 1600 pounds. The bulging between the rivets at that pressure was *one-third* of an inch; but still no part of the iron gave way. At 1625 pounds pressure the box burst, and in precisely the same way as in the first experiment — one of the stays drawing through the iron plate, and stripping the thread *in the plate.*

These experiments prove a number of facts of great value and importance to the engineer. In the first place, they show that, with regard to iron stay-bolts, their tensile strength is at least equal to the grip of the plate.

The grip of the copper bolt is evidently less. As each stay, in the first case, bore the pressure on an area of $5 \times 5 = 25$ square inches, and in the second, on an area of $4 \times 4 = 16$ square inches, the total strains borne by each stay were, for the first $815 \times 25 = 20{,}375$ pounds on each stay; and for the second $1625 \times 16 = 26{,}000$ pounds on each stay. These strains were less, however, than the tensile strength of the stays, which would be about 28,000 pounds.

The properly stayed surfaces are the strongest part of boilers when kept in good repair.

DEFINITIONS AS APPLIED TO BOILERS AND BOILER MATERIALS.

Tensile strength is the absolute resistance which a body makes to being torn apart by two forces acting in opposite directions.

Working Strength. — The term "working strength" implies a certain reduction made in the estimate of the strength of materials, so that, when the instrument or machine is put to use, it may be capable of resisting a greater strain than it is expected on the average to sustain.

Safe Working Pressure, or Safe Load. — The safe working pressure of steam-boilers is generally taken as $\frac{1}{5}$ of the bursting pressure, whatever that may be.

Elasticity is that quality which enables a body or boiler to return to its original form after having been distorted or stretched by some extreme force.

Internal Radius. — The internal radius is $\frac{1}{2}$ of the diameter less the thickness of the iron. To find the internal radius of a boiler, take $\frac{1}{2}$ of the external diameter and subtract the thickness of the iron.

Longitudinal Seams. — The seams which are parallel to the length of a boiler are called the longitudinal seams.

Curvilinear Seams. — The curvilinear seams of a boiler are those around the circumference.

TABLE

DEDUCTED FROM EXPERIMENTS ON IRON PLATES FOR STEAM-BOILERS, BY THE FRANKLIN INSTITUTE, PHILADELPHIA.

Iron boiler-plate was found to increase in tenacity as its temperature was raised, until it reached a temperature of 550° above the freezing-point, at which point its tenacity began to diminish.

At	32° to 80°	tenacity	is	56,000	lbs.	or one-seventh below its maximum.
"	570°	"	"	66,000	"	the maximum.
"	720°	"	"	55,000	"	the same nearly as at 30°.
"	1050°	"	"	32,000	"	nearly one-half the maximum.
"	1240°	"	"	22,000	"	nearly one-third the maximum.
"	1317°	"	"	9,000	"	nearly one-seventh the maximum.

It will be seen by the above table that if a boiler should

become overheated, by the accumulation of scale on some of its parts or an insufficiency of water, the iron would soon become reduced to less than one-half its strength.

TABLE

SHOWING THE RESULT OF EXPERIMENTS MADE ON DIFFERENT BRANDS OF BOILER IRON AT THE STEVENS INSTITUTE OF TECHNOLOGY, HOBOKEN, N. J.

Thirty-three experiments were made upon iron taken from the exploded steam-boiler of the ferry-boat Westfield. The following were the results:—

	Lbs. per sq. inch.
Average breaking weight	41,653
16 experiments made upon high grades of American boiler-plate.	
Average breaking weight	54,123
15 experiments made upon high grades of American flange-iron.	
Average breaking weight	42,144
6 experiments made upon English Bessemer steel.	
Average breaking weight	82,621
5 experiments made upon English lowmoor boiler-plate.	
Average breaking weight	58,984
6 experiments made upon samples of tank-iron from different manufacturers.	
Average breaking weight, No. 1	43,831
" " " No. 2	42,011
" " " No. 3	41,249
2 experiments made on iron taken from the exploded steam-boiler of the Red Jacket.	
Average breaking weight	49,000

It will be noticed that the above experiments reveal a great variation in the strength of boiler-plates of different grades of iron, and furnish conclusive evidence that the tensile strength of boiler-iron ought to be taken at 50,000 pounds to the square inch instead of 60,000.

FEED-WATER HEATERS.

Inattention to the temperature of feed-water for boilers is entirely too common; as the saving in fuel that may be effected by thoroughly heating the feed-water — by means of the exhaust steam in a properly constructed heater — would be immense, which will be seen from the following facts:

A pound of feed-water entering a steam-boiler at a temperature of 50° Fah., and evaporating into steam of 60 pounds pressure, requires as much heat as would raise 1157 pounds of water 1 degree. A pound of feed-water raised from 50° Fah. to 220° Fah., requires 987 thermal units of heat; which, if absorbed from exhaust-steam passing through a heater, would be a saving of 15 per cent. in fuel. Feed-water, at a temperature of 200° Fah., entering a boiler, as compared in point of economy with feed-water at 50°, would effect a saving of over 13 per cent. in fuel; and with a well-constructed heater there ought to be no trouble in raising the feed-water to a temperature of 212° Fah.

If we take the normal temperature of the feed-water at 60°, the temperature of the heated water at 212°, and the boiler pressure at 20 pounds, the total heat imparted to the steam in one case is $1192{\cdot}5° - 60° = 1132{\cdot}5°$, and in the other case $1192{\cdot}5° - 212° = 980{\cdot}5°$, the difference being 152°, or a saving of $\frac{152}{1132{\cdot}5} = 13{\cdot}4$ per cent.

Supposing the feed-water to enter the boiler at a temperature of 32° Fah., each pound of water will require about 1200 units of heat to convert it into steam, so that the boiler will evaporate between $6\frac{2}{3}$ and $7\frac{1}{2}$ pounds of water per pound of coal. The amount of heat required to convert a pound of water into steam varies with the pressure, as will be seen by the following table.

TABLE

SHOWING THE UNITS OF HEAT REQUIRED TO CONVERT ONE POUND OF WATER, AT THE TEMPERATURE OF 32° FAHR., INTO STEAM AT DIFFERENT PRESSURES.

Pressure of Steam in Pounds per sq. inch by Gauge.	Units of Heat.	Pressure of Steam in Pounds per sq. inch by Gauge.	Units of Heat.
1	1,148	110	1,187
10	1,155	120	1,189
20	1,161	130	1,190
30	1,165	140	1,192
40	1,169	150	1,193
50	1,173	160	1,195
60	1,176	170	1,196
70	1,178	180	1,198
80	1,181	190	1,199
90	1,183	200	1,200
100	1,185		

If the feed-water has any other temperature, the heat necessary to convert it into steam can easily be computed. Suppose, for instance, that its temperature is 65°, and that it is to be converted into steam having a pressure of 80 pounds per square inch. The difference between 65 and 32 is 33; and subtracting this from 1181 (the number of units of heat required for feed-water having a temperature of 32°), the remainder, 1148, is the number of units for feed-water with the given temperature.

Yet it must be understood that any design of heater that offers such resistance to the free escape of the exhaust steam as to neutralize the gain that would otherwise be obtained from its use, ought to be avoided, as the loss occasioned by back pressure on the exhaust in many instances overbalances that derived from the heating of the feed-water.

It is a common practice on steamships to heat the feed-

water to 135° or 140° before sending it into the boiler. Where the jet condenser is used, this extra heat is derived from the blow-water; but as this means of heating is not available with the surface condenser, it is generally derived from a water-jacket surrounding the smoke-stack, or a spiral pipe within the stack. But although any heat imparted to the feed-water is a clear gain, yet the cost, complication, and danger of these arrangements generally overbalance the benefits derived from their use.

STEAM-JACKETS.

The steam in passing from the boiler to the cylinder sustains a loss by condensation, friction, etc., more particularly if the pressure be high. The conducting properties of the metal rob the steam of its heat in proportion to the difference of the temperature. The office of the steam-jacket is to prevent this waste by keeping the walls of the cylinder at a constant temperature, so as to prevent the pressure of water in the cylinder and the resulting inconvenience.

The benefit to be derived from the use of the steam-jacket has not heretofore been fully understood, as an idea very generally prevailed among engineers, that waste by radiation was the only loss incident to the cooling of a steam-cylinder, and that this loss would be as great in the jacket as in the cylinder; whereas the loss is by no means measurable by the loss from the radiation, but is a much larger loss, and arises from the fact of the inner surface of the cylinder being cooled and heated by the steam at every stroke of the engine.

By comparing the diagrams obtained from cylinders without jackets with the theoretical curve, the loss has been found to amount in different cases from 10 to 15 per

cent. This loss is caused by the circumstance that the mass of the cylinder must remain at the average temperature intermediate between the highest and the lowest temperatures of the steam, so that when high-pressure steam, which also has a high temperature, enters the cylinder, a considerable quantity of it is at once condensed, owing to the abstraction of heat by the metal, and also to the transformation of a part of the heat into mechanical power. Hence, the necessity of clothing high-pressure cylinders and pipes with felt or other non-conducting substance to prevent the absorption of the caloric.

LOSS OF PRESSURE IN CYLINDERS INDUCED BY LONG STEAM-PIPES.

It is well known that the initial pressure in steam-cylinders seldom equals the boiler pressure. This loss of pressure is usually attributed to the frictional resistance of the steam-pipe and condensation within the latter. There is reason to believe, however, that although such a deduction is consistent with facts in many cases, it is by no means always so. It is of course quite possible to make a steam-pipe so small, and so full of bends and sharp turns, that it will cause considerable resistance, and consequently loss of power. But it would perhaps be found on investigation, in all cases where a considerable loss of power takes place, that the velocity of the steam is over 100 feet per second.

The only inducement to make steam-pipes too small is the first cost; but the wisdom of such economy is extremely doubtful, as, when steam-pipes are large enough, very little loss of pressure takes place, even though the pipe be two or three hundred yards long, so far as frictional resistance can affect the question.

There is only one other cause of loss of pressure, and that

is condensation. The remedy for this is obvious, as, if the steam-pipe be protected, the loss of pressure will be very slight. One of the best means to accomplish this is to lay the steam-pipe under ground, in large wooden troughs,—water-proof, if the ground be damp,—the troughs to be filled with dried saw-dust or fine dry sand. If this arrangement be inadmissible, then the pipes should be covered with felt, or some one or other of the various compositions in use for that purpose. The loss by condensation may in this manner be reduced to one or two per cent. of the whole quantity of steam used by the engine.

PRIMING IN STEAM-CYLINDERS.

Steam almost invariably contains more or less fine particles of water in its ordinary form; and, although every effort is made in steam-engineering to procure dry steam, or steam as free as possible from such particles, these efforts are not always successful, perhaps for the reason that the precise nature of this action is not understood, as it is of course not subject to inspection under such circumstances as occur in practice.

There are some reasons for supposing that the water, under the influence of some unknown law, ascends the side of the boiler in a thin sheet, and thus flows out with the steam from the steam-pipe; but there are stronger reasons for supposing it to rise in the form of thick spray mingled intimately with the vapor. This theory derives much support from the fact that simple deflecting plates of iron, placed in boilers in such a manner as to deflect the water and throw it against the sides of the boiler, to descend by its own gravity, have, in numerous instances, almost or entirely remedied this evil.

Some boilers are far more liable to work water than

others; and the reason cannot always be satisfactorily assigned for this difference; but in general a large area of water surface in the boiler, or, in other words, a liberal provision for the escape or disengagement of the steam from the water, so that it does not rise therefrom in any considerable velocity, tends very greatly to prevent priming. The steam-domes added on the top of many high-pressure boilers, and the steam chimney on low-pressure boilers, are both intended for the same purpose, to take the steam at a considerable elevation, so as to avoid the commotion of the water as far as possible.

The steam chimney, however, in the last-named example, contains the heated smoke stack or uptake, which tends very considerably to dry the steam by evaporating all the particles of water which come in contact with it. But with all precautions, engines are always liable to receive a greater or less quantity of water, to which may be added an allowance for the quantities, sometimes very considerable, which are condensed by contact with the cold metal of the cylinder in commencing to work.

As water is incompressible, except to a very small degree, and as the piston at each stroke comes into almost absolute contact with all parts of the cylinder end, it follows that a quantity of water sufficient to more than fill the small space remaining before the piston at the end of the stroke, must necessarily compel either a stoppage of the engine or a fracture of some portion of the machinery, unless means are provided for its escape.

OILS AND OILING.

Oils are divisible into two distinct classes — fat or fixed oils, and the essential or volatile oils. The former are usually bland to the taste; the latter, hot and pungent.

Whether of animal or vegetable origin, they possess the same ultimate constituents — carbon, hydrogen, and generally oxygen, and in nearly the same proportions. The fat oils are mixtures of three substances of similar properties; two of them at ordinary temperature are solid, called stearine and margarine, and the third is a fluid called oleine. The proportion of the latter gives softness and fluidity to the compound.

In close vessels, oils may be preserved fresh for a long time; but in contact with air they undergo progressive changes. Oils which thicken, and eventually dry into a transparent, flexible substance, are said to be drying or siccative, and used for preparations for varnishes and painters' colors. Other oils do not become dry, though they turn thick, become less combustible, and assume an offensive smell; they are then called rancid, and exhibit an acid reaction, which may be removed in a great measure by boiling the oil along with water and a little common magnesia for a quarter of an hour, or until it has lost the property of reddening litmus.

Now, as the cost of oil, like that of fuel, is among the heaviest items of expenditure incident to the use of the steam-engine, its reckless waste can only be attributed to ignorance; as no one, however careless, would hardly be guilty of such criminal waste in the use of oil as is often manifest, especially when any intelligent person can learn by a very short experience the proper quantity needed for any bearing, and that the amount of oil which a bearing will carry is limited, as, after the surface has been covered, every drop of oil poured on the journal or rubbing surface, is thrown off and wasted. The common practice of going about with a squirt-can and spirting oil at the bearings of a steam-engine, not into them, cannot be too severely censured.

The frequency with which oil is to be supplied into the cylinder is also a very important point, for the same rule applies here as to the bearing, as all that is not essential to the work is thrown away by the engine or carried out with the exhaust. It not unfrequently happens that exhaust-pipes have their areas greatly diminished in consequence of becoming coated with the tallow so needlessly poured into the cylinders.

On the question of lubricating steam-cylinders, like many others connected with steam-engineering, a difference of opinion exists among engineers, for while some claim that a lubricant is not at all needed, others stoutly contend that it is absolutely necessary. And in defence of the former theory, numerous instances might be cited of steam-engines that have been running for years without a drop of oil in their cylinders or steam-chests; but it must be borne in mind that these are exceptional cases, and may be due to the state of the steam, the construction of the engine, and nature of the metals in contact. By the state of the steam is meant whether it is saturated or superheated. Therefore, because one engine here and there runs without oil in the parts mentioned, it does not follow, as a rule, that no valve-seat or piston requires to lubricated.

It may be well to observe here that the waste of grease incurred by its lavished use is not the only loss occasioned, but the injury which the cylinder, valve-faces, and packing sustain from the too free use of tallow renders it highly detrimental to the durability of the parts mentioned. In the rendering of rough fats, such as are used for greasing cylinders and pistons, sulphuric acid is freely used. The quantity of the acid used amounts to, at the least, 12 per centum of the weight of fat, and it combines at once with the whole of the fatty matter; a portion of the acid is

removed by washing the grease in water at a high temperature; but a certain part remains behind and becomes a constituent of the rendered mass.

This acid is set free when introduced to the steam-cylinder by the heat therein, and, though necessarily small in quantity to the proportion of grease introduced, at once exerts its evil influence, and slowly but surely destroys the metal. The iron is eaten up, and the carbon alone remains. Animal fats are themselves acids, chemically speaking; but these are not specially injurious to iron. The best way to avoid the trouble before-mentioned is to use good refined tallow; but as it is somewhat difficult to procure such an article nowadays, lard-oil should be used, as it is undoubtedly the next best lubricant for the steam-cylinders.

TABLE

OF COEFFICIENTS OF FRICTIONS BETWEEN PLANE SURFACES.

Sliding surface.	Surface at rest.	State of the Surfaces.			Coefficient of Friction.
Cast-iron.	Wrought-iron.	Fibres of both surfaces parallel to motion.	Surfaces unctuous.		0.143
			Without lubric.		0.152
			Surfaces unctuous.		0.144
Cast-iron.	Cast-iron.	" "	Lubricated with	tallow.	0.100
				lard.	0.070
				olive-oil.	0.064
				lard and pl'bago.	0.055
Wrought-iron.	Bronze.	Fibres parallel to motion.	Without lubric.		0.072
			Surfaces unctuous.		0.060
			Lubricated with	tallow.	0.103
				lard.	0.075
				olive-oil.	0.078
Bronze.	Wrought-iron.	" "	Without lubric.		0.161
			Surfaces unctuous.		0.166
			Lubricated with	tallow.	0.081
				lard and pl'bago.	0.089
				olive-oil.	0.072

TABLE—(*Continued*)

OF COEFFICIENTS OF FRICTIONS BETWEEN PLANE SURFACES.

Sliding surface.	Surface at rest.	State of the Surfaces.		Coefficient of Friction.
Cast-iron.	Bronze.	" "	Without lubric.	0.147
			Surfaces unctuous.	0.132
			Lubricated with tallow.	0.103
			Lubricated with lard.	0.075
			Lubricated with olive-oil.	0.078
Bronze.	Cast-iron.	" "	Without lubric.	0.217
			Surfaces unctuous.	0.107
			Lubricated with tallow.	0.086
			Lubricated with olive-oil.	0.077
Bronze.	Bronze.	" "	Without lubric.	0.201
			Surfaces unctuous.	0.134
			Lubricated with olive-oil.	0.058
Brass.	Cast-iron.	" "	Without lubric.	0.189
			Surfaces unctuous.	0.115
			Lubricated with tallow.	0.072
			Lubricated with lard.	0.068
			Lubricated with olive-oil.	0.066
Steel.	Cast-iron.	" "	Without lubric.	0.202
			Lubricated with tallow.	0.105
			Lubricated with lard.	0.081
			Lubricated with olive-oil.	0.079
Steel. Steel.	Wrought-iron. Bronze.	Fibres of iron parallel to motion.	Lubricated with tallow.	0.093
			Lubricated with lard.	0.076
			Without lubric.	0.152
			Lubricated with tallow.	0.056
			Lubricated with olive-oil.	0.053
			Lubricated with lard and pl'bago.	0.076

What is the object of caulking?

Focus.—Focus in geometry is that point in the transverse axis of a conic section at which the double ordinate is equal to a perimeter, or to a third proportional to the transverse and conjugate axis.

GRATE-BARS.

The grate-bar has not heretofore received that consideration from engineers and steam users that its importance, in an economical point of view, so eminently deserves.

Perfect combustion is the starting-point in the generation of steam; the conversion of coal and air into heat must be the first process, and the second is to apply that heat with full effect to the boiler.

The oxygen of the air is the only supporter of combustion, and the rate of combustion produced, and the amount of heat generated in the furnace, depend on the quantity of air supplied, and the quantity of air admitted depends on the size of the opening through which it passes.

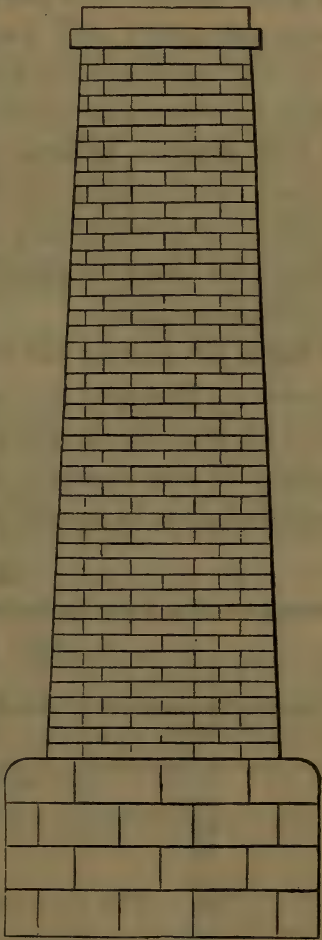
Then, as a matter of course, the grate-bars that offer the least obstruction to the air passing through them, and afford the largest area for the air combined with an equal distribution of the same, must be the most perfect for the purposes of combustion.

The destruction of grate-bars may be traced to three causes—breaking, warping, and burning out; consequently, grate-bars, to be durable and efficient, should have a narrow surface exposed to the fire, and the spaces for admitting the air should be numerous and well distributed.

The metal constituting the bar should be distributed in the best possible manner, to relieve the grate from all undue strain arising from unequal expansion and contraction; there should also be considerable depth, in order that the lower edges may keep cool and prevent the possibility of warping or twisting.

Grate-bars of good design and proportions are frequently ruined by being exposed to a white heat whenever a fresh fire is started, when, by distributing a thin layer of fresh coal over their surface before the shavings and wood are applied, they may be preserved intact for years.

CHIMNEYS.



The object of a chimney is to convey away the smoke and to produce a draught—that is, a current of fresh, dry air through the coals on the grate; this draught is produced by the difference in the specific gravity of the air inside and outside of the chimney. If the quality of the gases inside and outside were always the same, formulæ could be established for the size of chimneys with a considerable degree of accuracy.

The gases inside of a chimney are generally composed of atmospheric air, free nitrogen, carbonic acid, carbonic oxide, steam, free hydrogen, free carbon, sulphurous acid, and other elements. If the relative amount of these gases and their temperature were always the same, there would not be much difficulty in determining the proportions; but as these conditions are continually changing, as well by the gradual consumption of the coal on the grate, as by the management of the party in charge, it is almost impossible to arrive at any exact or definite conclusion. The air out-

side the chimney is also continually undergoing changes, produced by moisture, temperature, density, etc.

For stationary and marine boilers the chimneys are generally of a uniform height, arising from the nature of the structures with which they are connected, and hence the approximate amount of combustion on a square foot of grate surface, and the resulting evaporation of water per hour, are pretty well known from practical observations; but still, experiments are greatly needed to determine the proper proportions of chimneys for different kinds of fuel.

For marine boilers the general rule is to allow 14 square inches area of chimney for each nominal horse-power. For stationary boilers, the area of the chimney should be one-fifth greater than the combined area of all the flues or tubes.

Rule *for finding the Required Area of Chimney for any Boiler.*—Multiply the nominal horse-power of the boiler by 112, and divide the product by the square root of the height of the chimney in feet. The quotient will be the required area in square inches.

TABLE

SHOWING THE PROPER DIAMETER AND HEIGHT OF CHIMNEY FOR ANY KIND OF FUEL.

Nominal Horse-power of Boiler.	Height of Chimney in Feet.	Inside Diameter at Top.
10	60	1 foot 2 inches.
12	75	1 " 2 "
16	90	1 " 4 "
20	99	1 " 5 "
30	105	1 " 9 "
50	120	2 feet 2 "
70	120	2 " 6 "
90	120	2 " 10 "
120	135	3 " 2 "
160	150	3 " 7 "
200	165	3 " 11 "
250	180	4 " 4 "

SMOKE.

Notwithstanding the number of smoke-burning furnaces which have at different times been introduced, it cannot be said that any plan has yet been contrived which so far satisfies the conditions of the problem as to command general recognition of its suitability, or to lead to its general adoption.

These plans operate either on the principle of admitting air above the fuel to burn the smoke,—which has the radical defect that the production of smoke in ordinary furnaces is variable, whereas the admission of air is constant, so that either too much or too little will generally enter, — or on the principle of passing the smoke over the incandescent fuel, or through red-hot pipes or fire-brick passages, which, though it will diminish the smoke, will rarely wholly prevent it.

A great many experiments would be tried, but a lack of knowledge of the principles involved would, in a majority of cases, render success impossible, as not one engineer in a hundred, if required to consume smoke, so far as his own furnace is concerned, would have any clear idea how to proceed to do it.

From how many smoke-stacks throughout the land can great volumes of smoke, as black as midnight, be seen, at almost all times, rolling upward, carrying with them the most valuable proportions of the fuel! Each one of these advertises a great waste, which is generally produced by the boilers being too small.

With boilers of suitable proportions, grate surface adequate to the quantity of fuel to be consumed, and furnaces properly constructed, this waste occasioned by smoke would not occur; as the loss and inconvenience caused by smoke are generally aggravated by a false economy in

the first cost of boilers, a want of skill in their setting, and ignorance and carelessness in their management.

But it must be understood that all that comes out of the chimney is not smoke, by any means. Bituminous coal contains from five to six per cent. of hydrogen, which unites with the oxygen necessary to combustion, and makes water. A ton of bituminous coal will make nearly one-third of a ton of water, in the form of steam.

That this steam is black does not necessarily indicate the presence of much carbon, as a grain of soot, if distributed evenly in fine particles through a cubic foot of steam, would color it blacker than the ace of spades.

Now it requires no argument to show that this steam cannot be burned. It may be condensed by being made to pass through tubes kept at a low temperature, though a draught could only be maintained artificially under these conditions, but it cannot be consumed. If it were possible to separate the carbon atoms from the vapor in which they are held suspended, they could be burned; but such a separation could not be effected, and if it could, the amount of fuel saved would be very small.

Since the days of Watt, the consumption of smoke has attracted the attention of scientists, inventors, and engineers, but, so far, without any very practical results, as the methods that offered the most plausible solution of the problem involved in the burning of smoke have invariably failed to produce such results as would warrant their adoption into general use. A uniform supply of *fuel* to the furnace, and the introduction of *air* above the fire, were advocated as furnishing a remedy for the loss occasioned by smoke; but the former was, in most cases, found impracticable and inconvenient on account of the varying circumstances involved in the management of furnaces; whilst the latter was frequently productive of more waste

than that occasioned by smoke, in consequence of the current of cool air above the fire being constant, and the quantity of fuel on the grate and the temperature of the furnace seldom so.

When smoke is once formed, it cannot be burned by any known device.

The consumption of smoke would be a great benefit, not more so in point of economy than in comfort and convenience.

MENSURATION OF THE CIRCLE, CYLINDER, SPHERE, ETC.

1. **The circle contains a greater area** than any other plain figure bounded by an equal perimeter or outline.

2. The areas of circles are to each other as the squares of their diameters.

3. The diameter of a circle being 1, its circumference equals 3·1416.

4. The diameter of a circle is equal to ·31831 of its circumference.

5. The square of the diameter of a circle being 1, its area equals ·7854.

6. The square root of the area of a circle multiplied by 1·12837 equals its diameter.

7. The diameter of a circle multiplied by ·8862, or the circumference multiplied by ·2821, equals the side of a square of equal area.

8. The sum of the squares of half the chord and versed sine divided by the versed sine, the quotient equals the diameter of corresponding circle.

9. The chord of the whole arc of a circle taken from eight times the chord of half the arc, one-third of the remainder equals the length of the arc; or,

10. The number of degrees contained in the arc of a

circle, multiplied by the diameter of the circle and by ·008727, the product equals the length of the arc in equal terms of unity.

11. The length of the arc of a sector of a circle multiplied by its radius, equals twice the area of the sector.

12. The area of the segment of a circle equals the area of the sector, minus the area of a triangle whose vertex is the centre, and whose base equals the chord of the segment; or,

13. The area of a segment may be obtained by dividing the height of the segment by the diameter of the circle, and multiplying the corresponding tabular area by the square of the diameter.

14. The sum of the diameters of two concentric circles multiplied by their difference, and by ·7854, equals the area of the ring or space between them.

15. The sum of the thickness and internal diameter of a cylindric ring multiplied by the square of its thickness, and by 2·4674, equals its solidity.

16. **The circumference of a cylinder** multiplied by its length or height equals its convex surface.

17. The area of the end of a cylinder multiplied by its depth equals its cubical capacity.

18. The square of the diameter of a cylinder multiplied by its length, and divided by any other required length, the square root of the quotient equals the diameter of the other cylinder of equal contents or capacity.

19. **The square of the diameter of a sphere** multiplied by 3·1416 equals its convex surface.

20. The cube of the diameter of a sphere multiplied by ·5236 equals its solid contents.

21. The height of any spherical segment or zone multiplied by the diameter of the sphere of which it is a part,

and by 3·1416, equals the area or convex surface of the segment; or,

22. The height of the segment multiplied by the circumference of the sphere of which it is a part, equals the area.

23. The solidity of any spherical segment is equal to three times the square of the radius of its base, plus the square of its height, and multiplied by its height and by ·5236.

24. The solidity of a spherical zone equals the sum of the squares of the radii of its two ends, and one-third the square of its height multiplied by the height and by 1·5708.

25. The capacity of a cylinder 1 foot in diameter and 1 foot in length equals 5·875 of a United States gallon.

26. The capacity of a cylinder 1 inch in diameter and 1 inch in length equals ·0034 of a United States gallon.

27. The capacity of a sphere 1 foot in diameter equals 3·9156 United States gallons.

28. The capacity of a sphere 1 inch in diameter equals ·002165 of a United States gallon; hence,

29. The capacity of any other cylinder in United States gallons is obtained by multiplying the square of its diameter by its length, or the capacity of any other sphere by the cube of its diameter, and by the number of United States gallons contained as above in the unity of its measurement.

Rule *for finding the Area of a Circle.*—Multiply the circumference by one-quarter of the diameter; or, multiply the square of the diameter by ·7854; or, multiply the square of the circumference by ·07958; or, multiply half the circumference by half the diameter; or, multiply the square of half the diameter by 3·1416.

CENTRAL AND MECHANICAL FORCES AND DEFINITIONS.

Acceleration.—Acceleration is the increase of velocity in a moving body caused by the continued action of the motive force. When bodies in motion pass through equal spaces in equal time, or, in other words, when the velocity of the body is the same during the period that the body is in motion, it is termed uniform motion, of which we have a familiar instance in the motion of the hands of a clock over the face of it; but a more correct illustration is the revolution of the earth on its axis. In the case of a body moving through unequal spaces in equal times, or with a varying velocity, if the velocity increase with the duration of the motion, it is termed accelerated motion; but if it decrease with the duration of the motion, it is termed retarded motion.

Affinity.—Affinity is a term used in chemistry to denote that kind of attraction by which the particles of different bodies unite, and form a compound possessing properties distinct from any of the substances which compose it. Thus, when an acid and alkali combine, a new substance is formed called a salt, perfectly different in its chemical properties from either an acid or an alkali; and the tendency which these have to unite is said to be in consequence of affinity.

Angle.—If two lines drawn on a plane surface are so situated that they meet in a point, or would do so, if long enough, they form an opening, which is called an angle. The one line meeting another makes the angle on both sides equal to each other; then these angles are each called a right angle, and in this case the one line is said to be perpendicular to the other, or, in the language of mechanics, the one line is said to be square with the other; and if

the one line be horizontal, the perpendicular is said to be plumb to it. The arc which measures a right angle is the quarter of the whole circumference, or is a quadrant, and contains 90 degrees; any angle measured by an arc less than this is acute (sharp), and if by an arc greater than a quadrant, obtuse (blunt).

Axle. — An axle is a shaft supporting a wheel; the wheel may turn on the axle, or be fastened to it, and the axle turn on bearings. Axles are viewed as having certain relations to girders in principle. Girders generally have their two ends resting on two points of support, and the load is either located at fixed distances from the props, or dispersed over the whole surface of the axle; the wheels may be considered the props and the journals the loaded parts. It is found that the inclined surface of the wheel-tire given by coning ranges from 1 to 12 to 1 to 20; and, as a matter of course, the direct tendency of the wheel under a load is to descend that incline, so that every vertical blow which the wheels may receive is compounded of two forces, viz., the one to crush the wheels in the direction of their vertical plane, and the other to move the lower parts of the wheels together. It will be seen that these two forces have a direct tendency to bend the axle somewhere between the wheels.

Capillary Attraction. — Capillary attraction is the property observable in small tubes and porous substances, such as sponge, lamp-wicking, thread, etc., of raising oil, water, or other fluids above their natural level. Hence the application of this principle is applied for obtaining a continuous supply of lubricating fluids between rubbing and revolving surfaces in motion, by means of a siphon constructed of wickings, worsted, or some other substance, one end of which is immersed in oil, and the other inserted in the tube through which the fluid is to be conducted.

Centre of Gravity.—The forces with which all bodies tend to fall to the earth may be considered parallel; hence every body may be considered as acted on by a system of parallel forces, whose resultant may be found, and these forces, in all positions of the body act on the same points in the same vertical direction. There is, therefore, in every body a point through which the resultant always passes, in whatever position it is placed. This point is called the centre of gravity of the body. The centre of gravity of a uniform cylinder or prism is in its axis, and at the middle of its length; of a right cone or pyramid it is also in the axis, but at one-fourth of the height from the base.

Dynamics.—Dynamics is that branch of mechanics which treats of forces in motion producing power and work. It comprehends the action of all kinds of machinery, manual and animal labor, in the transformation of physical work.

Energy.—This term is used to denote work, but the sense of it conveys an idea of a different virtue, namely, that of activity or vigor, which is power. We say that a man has a great deal of energy when he can accomplish much work in a short time, which is virtue of power; but if he accomplishes the same quantity of work in a much longer time, we do not give him credit for much energy. The term energy, if employed at all, ought to be applied to power alone; but as we have the expressive term power for that function, it is better to dispense with the term energy in dynamics.

Force.—Force is the cause of motion or change of motion in material bodies. Every change of motion, viz., every change in the velocity of a body, must be regarded as the effect of a force. On the other hand, rest, or the invariability of the state of motion of a body, must not be attributed to the absence of forces, for opposite forces

destroy each other and produce no effect. The gravity with which a body falls to the ground still acts, though the body rests; but this action is counteracted by the solidity of the material upon which it reposes.

Forces that are balanced so as to produce rest are called statical forces, or pressures, to distinguish them from moving, deflecting, accelerating, or retarding forces; *i. e.*, such as are producing motion, or a change in the direction or velocity of motion. This distinction is wholly artificial, for the same force may act in any of these modes; it may sometimes be a statical and sometimes an accelerating force.

Force is any action which can be expressed simply by weight, and is distinguished by a great variety of terms, such as attraction, repulsion, gravity, pressure, tension, compression, cohesion, adhesion, resistance, inertia, strain, stress, strength, thrust, burden, load, squeeze, pull, push, pinch, punch, etc., all of which can be measured or expressed by weight without regard to motion, time, power, or work.

Focus.— Focus in geometry is that point in the transverse axis of a conic section at which the double ordinate is equal to the perimeter, or to a third proportional to the transverse and conjugate axis.

Friction.— Friction is the resistance occasioned to the motion of a body when pressed upon the surface of another body which does not partake of its motion. Under these circumstances, the surfaces in contact have a certain tendency to adhere. Not being perfectly smooth, the imperceptible asperities which may be supposed to exist on all surfaces, however highly polished, become to some extent interlocked, and, in consequence, a certain amount of force is requisite to overcome the mutual resistance to motion of the two surfaces and to maintain the

42

sliding motion even when it has been produced. By increasing the pressure, the resistance to motion is increased also; and on the other hand, by rendering the surfaces more smooth and by lubrication, its amount is greatly diminished, but can never be entirely annulled.

Friction cannot be strictly called a force, unless that term be taken in a negative sense. The tendency of force, in the rigid meaning of the word, is to produce motion; whereas the tendency of friction is to destroy motion.

Friction Rollers.—The obstruction which a cylinder meets in rolling along a smooth plane is quite distinct in its character, and far inferior in its amount to that which is produced by the friction of the same cylinder drawn lengthwise along a plane. For example, in the case of wood rolling on wood, the resistance is to the pressure, if the cylinder be small, as 16 or 18 to 1000, and if the cylinder be large, this may be reduced to 6 to 1000. The friction from sliding, in the same cases, would be to the pressure as 2 to 10 or 3 to 10, according to the nature of the wood. Hence, by causing one body to roll on another, the resistance is diminished from 12 to 20 times. It is therefore a principle, in the composition of machines, that attrition should be avoided as much as possible, and rolling motions substituted whenever circumstances admit.

Gravity and Gravitation.—These terms are often used synonymously to denote the mutual tendency which all bodies in nature have to approach each other.

Gravity, Specific.—The specific gravity of a body is the ratio of its weight to an equal volume of some other body assumed as a conventional standard. The standard usually adopted for solids and liquids is rain or distilled water at a common temperature.

In bodies of equal magnitudes, the specific gravities are directly as the weights or as their densities. In bodies of

the same specific gravities, the weights will be as the magnitudes. In bodies of equal weights the specific gravities are inversely as the magnitudes. The weights of different bodies are to each other in the compound ratio of their magnitudes and specific gravities. Hence, it is obvious that of the magnitude weight and specific gravity of a body, any two of these being given, the third may be found.

A body immersed in a fluid will sink if its specific gravity be greater than that of the fluid; if it be less, the body will rise to the top, and be only partly immerged; and if the specific gravity of the body and fluid be equal, it will remain at rest in any part of the fluid in which it may be placed. When a body is heavier than a fluid, it loses as much of its weight when immersed as is equal to a quantity of the fluid of the same bulk or magnitude.

If the specific gravity of the fluid be greater than that of the body, then the quantity of fluid displaced by the part immerged is equal to the weight of the whole body. And hence, as the specific gravity of the fluid is to that of the body, so is the whole magnitude of the body to the part immerged. The specific gravities of equal solids are as their parts immerged in the same fluid.

Gyration, the Centre of.—The centre of gyration is that point in which, if all the matter contained in a revolving system were collected, the same angular velocity will be generated in the same time by a given force acting at any place as would be generated by the same force acting similarly in the body or system itself. The distance of the centre of gyration from the point of suspension or the axis of motion, is a mean proportional between the distances of the centres of oscillation and gravity from the same point or axle.

Horse-power, or Power of a Horse.—The power of a

horse when applied to draw loads, as well as when made the standard of comparison for determining the value of other powers, has been variously stated. The relative strength of men and horses depends of course upon the manner in which their strength is applied. Thus, the worst way of applying the strength of a horse is to make him carry a weight up a steep hill, while the organization of the man fits him very well for that kind of labor. Three men climbing up a steep hill, each one having 100 pounds on his shoulder, will proceed faster than most horses with 300 pounds.

Hydrodynamics. — Hydrodynamics is that branch of general mechanics which treats of the equilibrium and motion of fluids. The terms hydrostatics and hydrodynamics have corresponding signification to the statics and dynamics in the mechanics of solid bodies, viz., hydrostatics is that division of the science which treats of the equilibrium of fluids, and hydrodynamics that which relates to their forces and motion. It is, however, very usual to include the whole doctrine of the mechanics of fluids under the general term of hydrodynamics, and to denote the divisions relative to their equilibrium and motion by the terms hydrostatics and hydraulics.

Hyperbola. — A plane figure formed by cutting a section from a cone by a plane parallel to its axis, or to any plane within the cone, which passes through the cone's vertex. The curve of the hyperbola is such that the difference between the distances of any point in it from two given points is always equal to a given right line.

If the vertexes of two cones meet each other so that their axes form one continuous straight line, and the plane of the hyperbola cut from one of the cones be continued, it will cut the other cone, and form what is called the opposite hyperbola, equal and similar to the former, and the

distance between the vertexes of the two hyperbolas is called the major axis, or transverse diameter. If the distance between a certain point within the hyperbola, called the focus, and any point in the curve be subtracted from the distance of said point in the curve from the focus of the opposite hyperbola, the remainder will always be equal to a given quantity, that is, to the major axis; and the distance of either focus from the centre of the major axis is called the eccentricity. The line passing through the centre, perpendicular to the major axis, and having the distance of its extremities from those of the axis equal to the eccentricity, is called the minor axis, or conjugate diameter. An ordinate to the major axis, a double ordinate, and an absciss mean the same as the corresponding lines in the parabola.

Impact.— The single instantaneous blow or stroke communicated from one body in motion to another either in motion or at rest.

Impenetrability.— In physics, one of the essential properties of matter, or body. It is a property inferred from invariable experience, and resting on this incontrovertible fact, that no two bodies can occupy the same portion of space in the same instant of time.

Impenetrability, as respects solid bodies, requires no proof: it is obvious to the touch. With regard to liquids, the property may be proved by very simple experiments. Let a vessel be filled to the brim with water, and a solid, incapable of solution in water, be plunged into it; a portion of the water will overflow exactly equal in bulk to the body immersed. If a cork be rammed hard into the neck of a vial full of water, the vial will burst, while its neck remains entire.

The disposition of air to resist penetration may be illustrated in the following way: Let a tall glass vessel

be nearly filled with water, on the surface of which a lighted taper is set to float; if over this glass a smaller cylindrical vessel, likewise of glass, be inverted and pressed downwards, the contained air maintaining its place, the internal body of the water will descend while the rest will rise up at the sides, and the taper will continue to burn for some seconds encompassed by the whole mass of liquid.

Impetus. — Impetus is the product of the mass and velocity of a moving body, considered as instantaneous, in distinction from momentum, with reference to time, and force, with reference to capacity of continuing its motion. Impetus in gunnery is the altitude through which a heavy body must fall to acquire a velocity equal to that with which the ball is discharged from the piece.

Incidence. — The term incidence in mechanics is used to denote the direction in which a body or ray of light strikes another body, and is otherwise called inclination. In moving bodies their incidence is said to be perpendicular or oblique according as their lines of motion make a straight line or an angle at the point of contact.

Inclination. — Inclination denotes the mutual approach or tendency of two bodies, lines, or planes towards each other, so that the lines of their direction make at the point of contact an angle of greater or less magnitude.

Inclined Plane. — An inclined plane is one of the mechanical powers; a plane which forms an angle with the horizon. The force which accelerates the motion of a heavy body on an inclined plane, is to the force of gravity as the sine of the inclination of the plane to the radius, or, as the height of the plane to its length.

Inertia. — Inertia is that property of matter by which it tends when at rest to remain so, and when in motion to continue in motion.

Levers.— Levers are classified into three different kinds or orders. When the fulcrum is between the force and the weight, the lever is called a lever of the first order; when the weight is between the force and the fulcrum, the lever is of the second order; when the force is between the weight and the fulcrum, the lever is of the third order. The levers of safety-valves for steam-boilers belong to this latter class.

Machines.—Machines are instruments employed to regulate motion so as to save either time or force. The maximum effect of machines is the greatest effect which can be produced by them. In all machines that work with a uniform motion there is a certain velocity, and a certain load of resistance that yields the greatest effect, and which are therefore more advantageous than any other.

A machine may be so heavily charged that the motion resulting from the application of any given power will be but just sufficient to overcome it, and if any motion ensue, it will be very trifling, and therefore the whole effect very small.

And if the machine is very lightly loaded, it may give great velocity to the load; but from the smallness of its quantity, the effect may still be very inconsiderable, consequently between these two loads there must be some intermediate one that will render the effect the greatest possible. This is equally true in the application of animal strength as in machines. The maximum effect of a machine is produced when the weight or resistance to be overcome is four-ninths of that which the power, when fully exerted, is able to balance, or of that resistance which is necessary to reduce the machine to rest, and the velocity of the part of the machine to which the power is applied should be one-third of the greatest velocity of the power.

The moving power and the resistance being both given, if the machine be so constructed that the velocity of the point to which the power is applied be to the velocity of the point to which the resistance is applied as four times the resistance to nine times the power, the machine will work to the greatest possible advantage.

This is equally true when applied to the strength of animals; that is, a man, horse, or other animal, will do the greatest quantity of work, by continued labor, when his strength is opposed to a resistance equal to four-ninths of his natural strength, and his velocity equal to one-third of his greatest velocity when not impeded.

Mass.—Mass is the real quantity of matter in a body, and is proportioned to weight when compared in one or the same locality. Mass is a constant quantity, whilst weight varies with the force of gravity which produces it.

Matter.—Matter is that of which bodies are composed, and occupies space. Matter is recognized as substance in contradistinction from geometrical quantities and physical phenomena, such as color, shadow, light, heat, electricity, and magnetism.

Mechanical Powers.—Mechanical powers are usually denominated the lever, inclined plane, wheel and axle, pulley, screw, and wedge.

The wheel and axle is, however, a revolving lever, the screw a revolving inclined plane, and the wedge a double inclined plane, thus reducing them to three in number, viz., lever, inclined plane, and pulley.

All these machines act on the same fundamental principle of vertical velocities; accordingly, the weight multiplied into the space it moves through is equal to the power multiplied into the space it moves through.

In all machines a portion of the effect is lost in overcoming the friction of the working parts; but in making

calculations upon them, it is made first as though no friction existed, a deduction being afterwards made.

Rules *for Finding the Effects of the Mechanical Powers.*

Inclined Plane.—As the length of the plane is to its height, so is the weight to the power.

Lever.— When the fulcrum (or support) of the lever is between the weight and the power, divide the weight to be raised by the power, and the quotient is the difference of leverage, or the distance from the fulcrum at which the power supports the weight. Or, multiply the weight by its distance from the fulcrum, and the power by its distance from the same point, and the weight and power will be to each other as their products.

When the fulcrum is at one extremity of the lever, and the power, or the weight, at the other. As the distance between the power, or weight, and fulcrum is to the distance between the weight, or power, and fulcrum, so is the effect to the power or the power to the effect.

Screw. — As the screw is an inclined plane wound round a cylinder, the length of the plane is found by adding the square of the circumference of the screw to the square of the distance between the threads, and taking the square root of the sum and the height is the distance between the consecutive threads.

Wedge.—When two bodies are forced from one another in a direction parallel to the back of the wedge. As the length of the wedge is to half its back, so is the resistance to the force.

Wheel and Axle.— The power multiplied by the radius of the wheel is equal to the weight multiplied by the radius of the axle; as the radius of the wheel is to the radius of the axle, so is the effect to the power.

When a series of wheels and axles act upon each other, either by belts or teeth, the weight or velocity will be

to the power or unity as the product of the radii, or circumferences of the wheels, to the product of the radii or circumferences of the axles.

Mechanics.— Mechanics is that branch of natural philosophy which treats of the three simple physical elements, force, motion, and time, with their combinations, constituting power, space, and work.

Modulus.— The modulus of the elasticity of any substance is a column of the same substance capable of producing a pressure on its base, which is to the weight causing a certain degree of compression as the length of the substance is to the diminution of its length.

Momentum. — Momentum, in mechanics, is the same with impetus or quantity of motion, and is generally estimated by the product of the velocity and mass of the body. This is a subject which has led to various controversies between philosophers, some estimating it by the mass into the velocity as stated above, while others maintain that it varies as the mass into the square of the velocity. But this difference seems to have arisen rather from a misconception of the term, than from any other cause. Those who maintain the former doctrine, understanding momentum to signify the momentary impact, and the latter as the sum of all the impulses, tell the motion of the body is destroyed.

Motion. — Motion, in mechanics, is a change of place, or it is that affection of matter by which it passes from one point of space to another. Motion is of various kinds, as follows:

Absolute motion is the absolute change of place in a moving body independent of any other motion whatever; in which general sense, however, it never falls under our observation.

All those motions which we consider as absolute, are in

fact only relative, being referred to the earth, which is itself in motion. By absolute motion, therefore, we must only understand that which is so with regard to some fixed point upon the earth, this being the sense in which it is delivered by writers on this subject.

Accelerated motion is that which is continually receiving constant accessions of velocity.

Angular motion is the motion of a body as referred to a centre, about which it revolves.

Compound motion is that which is produced by two or more powers acting in different directions.

Uniform motion is when a body moves continually with the same velocity, passing over equal spaces in equal times.

Natural motion is that which is natural to bodies or that which arises from the action of gravity.

Relative motion is the change of relative place in one or more moving bodies.

Retarded motion is that which suffers continual diminution of velocity, the laws of which are the reverse of those for accelerated motion.

Oscillation, Centre of.—The centre of oscillation is that point in a vibrating body into which, if the whole were concentrated and attached to the same axis of motion, it would vibrate in the same time the body does in its natural state. The centre of oscillation is situated in a right line passing through the centre of gravity, and perpendicular to the axis of motion.

Parallel Motions.—Contrivances of this kind are required for the conversion of rotary and alternating angular motion into rectilineal motion, and the converse; but the absolute necessity there is of guiding the path of a piston in a steam-engine has called forth more attention to the principles and mechanism of parallel motions than

would otherwise, in all probability, have been awarded to the subject for other purposes.

Motion is expressed by the following terms: Move, going, walking, passing, transit, involution and evolution, run, locomotion, flux, rolling, flow, sweep, wander, shift, flight, current, etc.

Pendulum. — If any heavy body, suspended by an inflexible rod from a fixed point, be drawn aside from the vertical position, and then let fall, it will descend in the arc of a circle, of which the point of suspension is the centre. On reaching the vertical position, it will have acquired a velocity equal to that which it would have acquired by falling vertically through the versed sine of the arc it has described, in consequence of which it will continue to move in the same arc until the whole velocity is destroyed; and if no other force than gravity acted, this would take place when the body reached a height on the opposite side of the vertical equal to the height from which it fell.

Having reached this height, it would again descend, and so continue to vibrate forever; but in consequence of the friction of the axis and the resistance of the air, each successive excursion will be diminished, and the body soon be brought to rest in the vertical position. A body thus suspended and caused to vibrate is called a pendulum; and the passage from the greatest distance from the vertical on the one side to the greatest distance on the other is called an oscillation.

Percussion. — The centre of percussion is that point in a body revolving about an axis at which, if it struck an immovable obstacle, all its motion would be destroyed, or it would not incline either way.

When an oscillating body vibrates with a given angular velocity, and strikes an obstacle, the effect of the impact

will be the greatest if it be made at the centre of percussion. For in this case the obstacle receives the whole revolving motion of the body; whereas, if the blow be struck in any other point, a part of the motion will be employed in endeavoring to continue the rotation.

Perpetual Motion.—In mechanics, a machine which, when set in motion, would continue to move forever, or, at least, until destroyed by the friction of its parts, without the aid of any exterior cause. The discovery of perpetual motion has always been a celebrated problem in mechanics, on which many ingenious, though in general ill-instructed, persons have consumed their time; but all the labor bestowed on it has proved abortive. In fact, the impossibility of its existence has been fully demonstrated from the known laws of matter. In speaking of perpetual motion, it is to be understood that, from among the forces by which motion may be produced, we are to exclude not only air and water, but other natural agents, as heat, atmospheric changes, etc. The only admissible agents are the inertia of matter, and its attractive forces, which may all be considered of the same kind as gravitation.

It is an admitted principle in philosophy, that action and reaction are equal, and that, when motion is communicated from one body to another, the first loses just as much as is gained by the second. But every moving body is continually retarded by two passive forces,—the resistance of the air and friction. In order, therefore, that motion may be continued without diminution, one of two things is necessary—either that it be maintained by an exterior force, (in which case it would cease to be what we understand by a perpetual motion,) or that the resistance of the air and friction be annihilated, which is practically impossible.

The motion cannot be perpetuated till these retarding

forces are compensated, and they can only be compensated by an exterior force, for the force communicated to any body cannot be greater than the generating force, and this is only sufficient to continue the same quantity of motion when there is no resistance. The error of confounding mere pressure with energy available to produce power is the main origin of the majority of attempts at perpetual motion, and even sometimes causes, among confused minds, exaggerating expectations about the effects to be obtained from mechanical contrivances. A wound-up spring is perfectly equivalent to a weight. It may exert a certain pressure, large in proportion to its size and strength; but unless it is allowed to unwind, it cannot produce motion or power. It is the same with compressed air or gases; they are in fact nothing but wound-up springs, with the difference, however, that, in place of needing mechanical power to wind them up, we may use either heat, chemical agencies, or electricity.

Pneumatics.—Pneumatics is the science which treats of the mechanical properties of elastic fluids, and particularly of atmospheric air. Elastic fluids are divided into two classes—permanent gases, and vapors. The gases cannot be converted into the liquid state by any known process of art, whereas the vapors are readily reduced to the liquid form by pressure or diminution of temperature. In respect of their mechanical properties, there is, however, no essential difference between the two classes.

Elastic fluids, in a state of equilibrium, are subject to the action of two forces, namely, gravity, and a molecular force acting from particle to particle.

Gravity acts on the gases in the same manner as on all other substances; but the action of the molecular forces is altogether different from that which takes place among the elementary particles of solids and liquids; for, in the

case of solid bodies, the molecules strongly attract each other, (whence results their cohesion,) and, in the case of liquids, exert a feeble or evanescent attraction, so as to be indifferent to internal motion; but, in the case of the gases, the molecular forces are repulsive, and the molecules, yielding to the action of these forces, tend incessantly to recede from each other, and, in fact, do recede until their further separation is prevented by an exterior obstacle.

Thus, air confined within a close vessel exerts a constant pressure against the interior surface, which is not sensible, only because it is balanced by the equal pressure of the atmosphere on the exterior surface. This pressure exerted by the air against the sides of a vessel within which it is confined is called its elasticity or elastic force or tension.

Power. — Power is the product of force and velocity; that is to say, a force multiplied by the velocity with which it is acting. The term horse-power is a unit of power, established by James Watt to be equivalent to a force of 33,000 pounds acting with a velocity of one foot per minute, or 150 pounds acting with a velocity of 220 feet per minute, which is the same as a force of 550 pounds acting with a velocity of one foot per second. Man-power is a unit of power established by Morin to be equivalent to 50 foot-pounds of power, or 50 effects; that is to say, a man turning a crank with a force of 50 pounds, and with a velocity of one foot per second, is a standard man-power.

Prime Movers. — Prime movers are those machines from which we obtain power, through their adaptation to the transformation of some available natural force into that kind of effort which develops mechanical power.

Statics is the science of forces in equilibrium. It embraces the strength of materials, of bridges, and of girders; the

stability of walls, steeples, and towers; the static momentum of levers, with their combination into weighing-scales, windlasses, pulleys, funicular machines, inclined planes, screws, catenaria, and all kinds of gearing.

Tools.—By the term tools, according to the definition given by Rennie, we understand instruments employed in the manual arts for facilitating mechanical operations by means of percussion, penetration, separation, and abrasion, of the substances operated upon, and for all which operations various motions are required to be imparted either to the tool or to the work.

Torsion.—Torsion, in mechanics, is the twisting or wrenching of a body by the exertion of a lateral force. If a slender rod of metal, suspended vertically, and having its upper end fixed, be twisted through a certain angle by a force acting in a plane perpendicular to its axis, it will, on the removal of the force, untwist itself, or return in the opposite direction with a greater or less velocity, and after a series of oscillations will come to rest in its original position.

The limits of torsion within which the body will return to its original state depend on its elasticity. A fine wire of a few feet in length may be twisted through several revolutions, without impairing its elasticity; and within those limits the force evolved is found to be perfectly regular, and directly proportional to the angular displacement from the position of rest. If the angular displacement exceeds a certain limit (as in a wire of lead, for example, before disruption takes place), the particles will assume a new arrangement, or take a set, and will not return to their original position on the withdrawal of the disturbing force.

Velocity.—Velocity is rate of motion. Velocity is independent of space and time, but in order to obtain its

value or expression as a quantity, we compare space with time. Thus, when the value of velocity of a moving body is required, we measure a space which the body passes through and divide that space with the time of passage, and the quotient is the velocity. Velocity, or rate of motion, is expressed by a variety of terms: speed, swiftness, rapidity, fleetness, speediness, quickness, haste, hurry, race, forced, march, gallop, trot, run, rush, scud, dash, spring, etc.

Weight.—The weight of a body is the force of attraction between the earth and that body. The weight of a body is greatest at the surface of the earth, and decreases above or below that surface. Above the surface, the weight decreases as the square of its distance from the centre of the earth, and below the surface the weight decreases simply as its distance from the centre.

Weights and Measures — The weights and measures of this country are identical with those of England. In both countries they repose, in fact, upon actually existing masses of metal (brass), which have been individually declared by law to be the units of the system. In scientific theory, they are supposed to rest upon a permanent and universal law of nature — the gravitation of distilled water at a certain temperature and under a certain atmospheric pressure.

In this aspect, the origination is with the grains, which must be such that 252,458 of these units of brass will be in just equilibrium with a cubic inch of distilled water; when the mercury stands at 30 inches in a barometer, and in a thermometer of Fahrenheit at 62 degrees, both for the air and for the water. Unfortunately, the expounders of this theory in England used only the generic term brass, and failed to define the specific gravity of the metal to be employed; the consequence of this omission is to leave room for an error of $\frac{1}{100000}$ in every attempt to reproduce

or compare the results. This is the minimum possible error; the maximum would be a fraction of the difference in specific gravity between the heaviest and lightest brass that can be cast.

Work. — Work is force acting through space, and is measured by multiplying the measure of the force by the measure of the space. Work is said to be performed when a pressure is exerted upon a body, and the body is thereby moved through space.

Work done is expressed by the following terms: hauled, dragged, raised, heaved, tilted, broken, crushed, thrown, wrought, fermented, labored, etc., or any expression which implies the three simple elements of force, velocity, and time. Power multiplied by the time of action is work; work divided by time is power. If work was independent of time, then any amount of work could be accomplished in no time. The greatest amount of work known to have been accomplished in the shortest time is that in the explosion of nitro-glycerine, which is instantaneous to our perception; but it required time, notwithstanding.

Workmanday.— A laborer working eight hours per day can exert a power of 50 foot-pounds. A day's work will then be $50 \times 8 \times 60 \times 60 = 1{,}440{,}000$ foot-pounds of work, which may be termed a workmanday. All kinds of heavy work can be estimated in workmandays, such as the building of a house, a bridge, a steamboat, canal and railroad excavations and embankments, loading or unloading a ship, powder and steam-boiler explosions, and the capability of heavy ordnance, etc.

The magnitude of the unit workmanday is easily conceived, because it is that amount of work which a laborer can accomplish in one day. Work expressed in foot-pounds, divided by 1,440,000, gives the work in workmandays.

THE CIRCLE.

The area of any circle is equal to the square of its diameter multiplied by ·7854; it is also equal to its circumference multiplied by half its radius. On the principle of the area of a triangle being equal to its base, multiplied by half its perpendicular height, a circle may be considered as composed of a great many triangles, whose bases are the circumference of the circle, and whose vertices are coincident with the centre of the circle.

A TABLE

CONTAINING THE DIAMETERS, CIRCUMFERENCES, AND AREAS OF CIRCLES, AND THE CONTENTS OF EACH IN GALLONS, AT 1 FOOT IN DEPTH. UTILITY OF THE TABLE.

EXAMPLES.

1. Required the circumference of a circle, the diameter being five inches?

In the column opposite the given diameter stands 15·708* inches, the circumference required.

2. Required the capacity in gallons of a can, the diameter being 6 feet and depth 10 feet?

In the fourth column from the given diameter stands 211·4472,* being the contents of a can 6 feet in diameter and 1 foot in depth, which being multiplied by 10 gives the required contents, 2114½ gallons.

3. Any of the areas in feet multiplied by ·03704, the product equals the number of cubic yards at 1 foot in depth.

4. The area of a circle in inches multiplied by the length or thickness in inches, and by .263, the product equals the weight in pounds of cast-iron.

*For decimal equivalents to the fractional parts of a gallon or an inch, see table on page 294.

TABLE

OF DIAMETERS, CIRCUMFERENCES, AND AREAS OF CIRCLES, AND THE CONTENTS IN GALLONS AT 1 FOOT IN DEPTH.

DIAM.	CIR.	AREA.	GALLONS.	DIAM.	CIR.	AREA.	GALLONS.
Inch.	Inch.	Inch.		Inch.	Inch.	Inch.	
1	3.1416	.7854	.04084	⅛	19.242	29.464	1.53213
⅛	3.5343	.9940	.05169	¼	19.635	30.679	1.59531
¼	3.9270	1.2271	.06380	⅜	20.027	31.919	1.65979
⅜	4.3197	1.4848	.07717	½	20.420	33.183	1.72552
½	4.7124	1.7671	.09188	⅝	20.813	34.471	1.79249
⅝	5.1051	2.0739	.10784	¾	21.205	35.784	1.86077
¾	5.4978	2.4052	.12506	⅞	21.598	37.122	1.93034
⅞	5.8905	2.7611	.14357	7	21.991	38.484	2.00117
2	6.2832	3.1416	.16333	⅛	22.383	39.871	2.07329
⅛	6.6759	3.5465	.18439	¼	22.776	41.282	2.14666
¼	7.0686	3.9760	.20675	⅜	23.169	42.718	2.22134
⅜	7.4613	4.4302	.23036	½	23.562	44.178	2.29726
½	7.8540	4.9087	.25522	⅝	23.954	45.663	2.37448
⅝	8.2467	5.4119	.28142	¾	24.347	47.173	2.45299
¾	8.6394	5.9395	.30883	⅞	24.740	48.707	2.53276
⅞	9.0321	6.4918	.33753	8	25.132	50.265	2.61378
3	9.4248	7.0686	.36754	⅛	25.515	51.848	2.69609
⅛	9.8175	7.6699	.39879	¼	25.918	53.456	2.77971
¼	10.210	8.2957	.43134	⅜	26.310	55.088	2.86458
⅜	10.602	8.9462	.46519	½	26.703	56.745	2.95074
½	10.995	9.6211	.50029	⅝	27.096	58.426	3.03815
⅝	11.388	10.320	.53664	¾	27.489	60.132	3.12686
¾	11.781	11.044	.57429	⅞	27.881	61.862	3.21682
⅞	12.173	11.793	.61324	9	28.274	63.617	3.30808
4	12.566	12.566	.65343	⅛	28.667	65.396	3.40059
⅛	12.959	13.364	.69493	¼	29.059	67.200	3.49440
¼	13.351	14.186	.73767	⅜	29.452	69.029	3.58951
⅜	13.744	15.033	.78172	½	29.845	70.882	3.68586
½	14.137	15.904	.82701	⅝	30.237	72.759	3.78347
⅝	14.529	16.800	.87360	¾	30.630	74.662	3.88242
¾	14.922	17.720	.92144	⅞	31.023	76.588	3.98258
⅞	15.315	18.665	.97058	10	31.416	78.540	4.08408
5	15.708	19.635	1.02102	⅛	31.808	80.515	4.18678
⅛	16.100	20.629	1.07271	¼	32.201	82.516	4.29083
¼	16.493	21.647	1.12564	⅜	32.594	84.540	4.39608
⅜	16.886	22.690	1.17988	½	32.986	86.590	4.50268
½	17.278	23.758	1.23542	⅝	33.379	88.664	4.61053
⅝	17.671	24.850	1.29220	¾	33.772	90.762	4.71962
¾	18.064	25.967	1.35028	⅞	34.164	92.885	4.82846
⅞	18.457	27.108	1.40962	11	34.557	95.033	4.94172
6	18.849	28.274	1.47025	⅛	34.950	97.205	5.05466

TABLE—(*Continued*)

OF DIAMETERS, CIRCUMFERENCES, AND AREAS OF CIRCLES, AND THE CONTENTS IN GALLONS AT 1 FOOT IN DEPTH.

Diam.	Cir.	Area.	Gallons.	Diam.	Cir.	Area.	Gallons.
Inch.	Inch.	Inch.		Inch.	Inch.	Inch.	
¼	35.343	99.402	5.16890	3 10	12 5½	11.5409	86.3074
⅜	35.735	101.623	5.28439	3 11	12	12.0481	90.1004
½	36.128	103.869	5.40119	4	12 6¾	12.5664	93.9754
⅝	36.521	106.139	5.51923	4 1	12 9⅞	13.0952	97.9310
¾	36.913	108.434	5.63857	4 2	13 1	13.6353	101.9701
⅞	37.306	110.753	5.75916	4 3	13 4⅛	14.1862	103.0300
				4 4	13 7¼	14.7479	110.2907
Ft. In.	Ft. In.	Feet.		4 5	13 10½	15.3206	114.5735
1	3 1⅝	.7854	5.8735	4 6	14 1⅝	15.9043	118.9386
1 1	3 4⅝	.9217	6.8928	4 7	14 4⅝	16.4986	123.3830
1 2	3 8	1.0690	7.9944	4 8	14 7⅞	17.1041	127.9112
1 3	3 11	1.2271	9.1766	4 9	14 11	17.7205	132.5209
1 4	4 2⅛	1.3962	10.4413	4 10	15 2⅛	18.3476	137.2105
1 5	4 5⅜	1.5761	11.7866	4 11	15 5¼	18.9858	142.0582
1 6	4 8½	1.7671	13.2150	5	15 8½	19.6350	146.8384
1 7	4 11⅝	1.9689	14.7241	5 1	15 11⅝	20.2947	151.7718
1 8	5 2¾	2.1816	16.3148	5 2	16 2¾	20.9656	156.7891
1 9	5 5⅞	2.4052	17.9870	5 3	16 5¾	21.6475	161.8886
1 10	5 9	2.6398	19.7414	5 4	16 9	22.3400	167.0674
1 11	6 2¼	2.8852	21.4830	5 5	17 0⅛	23.0437	172.3300
2	6 3⅜	3.1416	23.4940	5 6	17 3¼	23.7583	177.6740
2 1	6 6½	3.4087	25.4916	5 7	17 6⅜	24.4835	183.0973
2 2	6 9⅝	3.6869	27.5720	5 8	17 9⅝	25.2199	188.6045
2 3	7 0¾	3.9760	29.7340	5 9	18 0¾	25.9672	194.1930
2 4	7 3⅞	4.2760	32.6976	5 10	18 3⅞	26.7251	199.8610
2 5	7 7	4.5869	34.3027	5 11	18 7⅛	27.4943	205.6133
2 6	7 10¼	4.9087	36.7092	6	18 10⅛	28.2744	211.4472
2 7	8 1⅜	5.2413	39.1964	6 3	19 7½	30.6796	229.4342
2 8	8 4½	5.5850	41.7668	6 6	20 4⅞	33.1831	248.1564
2 9	8 7⅝	5.9395	44.4179	6 9	21 2⅜	35.7847	267.6122
2 10	8 10¾	6.3049	47.1505	7	21 11⅞	38.4846	287.8032
2 11	9 1⅞	6.6813	49.9654	7 3	22 9¼	41.2825	308.7270
3	9 5	7.0686	52.8618	7 6	23 6¾	44.1787	330.3859
3 1	9 8¼	7.4666	55.8382	7 9	24 4⅛	47.1730	352.7665
3 2	9 11⅜	7.8757	58.8976	8	25 1½	50.2656	375.9062
3 3	10 2½	8.2957	62.0386	8 3	25 11	53.4562	399.7668
3 4	10 5⅝	8.7265	65.2602	8 6	26 8⅜	56.7451	424.3625
3 5	10 8¾	9.1683	68.5193	8 9	27 5¾	60.1321	449.2118
3 6	10 11⅞	9.6211	73.1504	9	28 3¼	63.6174	475.7563
3 7	11 3	10.0846	75.4166	9 3	29 0⅝	67.2007	502.5536
3 8	11 6⅛	10.5591	78.9652	9 6	29 10⅛	70.8823	530.0861
3 9	11 9⅜	11.0446	82.5959	9 9	30 7½	74.6620	558.3522

TABLE—(*Concluded*)

OF DIAMETERS, CIRCUMFERENCES, AND AREAS OF CIRCLES, AND THE CONTENTS IN GALLONS AT 1 FOOT IN DEPTH.

DIAM.		CIR.		AREA.	GALLONS.	DIAM.		CIR.		AREA.	GALLONS.
Ft.	In.	Ft.	In.	Feet.		Ft.	In.	Ft.	In.	Feet.	
10		31	5	78.5400	587.3534	20	6	64	$4\frac{3}{4}$	330.0643	2468.3528
10	3	32	$2\frac{3}{8}$	82.5160	617.0876	20	9	65	$2\frac{1}{4}$	338.1637	2528.9233
10	6	32	$11\frac{3}{4}$	86.5903	647.5568	21		65	$11\frac{5}{8}$	346.3614	2590.2290
10	9	33	$9\frac{1}{4}$	90.7627	678.2797	21	3	66	9	354.6571	2652.2532
11		34	$6\frac{5}{8}$	95.0334	710.6977	21	6	67	$6\frac{1}{2}$	363.0511	2715.0413
11	3	35	$4\frac{1}{8}$	99.4021	743.3686	21	9	68	$3\frac{7}{8}$	371.5432	2778.5486
11	6	36	$1\frac{1}{2}$	103.8691	776.7746	22		69	$1\frac{3}{8}$	380.1336	2842.7910
11	9	36	$10\frac{7}{8}$	108.4342	810.9143	22	3	69	$10\frac{3}{4}$	388.8220	2907.7664
12		37	$8\frac{3}{8}$	113.0976	848.1890	22	6	70	$8\frac{1}{4}$	397.6087	2973.4889
12	3	38	$5\frac{3}{4}$	117.8590	881.3966	22	9	71	$5\frac{5}{8}$	406.4935	3039.9209
12	6	39	$3\frac{1}{4}$	122.7187	917.7395	23		72	3	415.4766	3107.1001
12	9	40	$0\frac{5}{8}$	127.6765	954.8159	23	3	73	$0\frac{1}{2}$	424.5577	3175.0122
13		40	10	132.7326	992.6274	23	6	73	$9\frac{7}{8}$	433.7371	3243.6595
13	3	41	$7\frac{1}{2}$	137.8867	1031.1719	23	9	74	$7\frac{1}{4}$	443.0146	3313.0403
13	6	42	$4\frac{7}{8}$	143.1391	1070.4514	24		75	$4\frac{3}{4}$	452.3904	3383.1563
13	9	43	$2\frac{1}{4}$	148.4896	1108.0645	24	3	76	$2\frac{1}{8}$	461.8642	3454.0051
14		43	$11\frac{3}{4}$	153.9384	1151.2129	24	6	76	$11\frac{5}{8}$	471.4363	3525.5929
14	3	44	$9\frac{1}{8}$	159.4852	1192.6940	24	9	77	9	481.1065	3597.9068
14	6	45	$6\frac{5}{8}$	165.1303	1234.9104	25		78	$6\frac{3}{8}$	490.8750	3670.9596
14	9	46	4	170.8735	1277.8615	25	3	79	$3\frac{7}{8}$	500.7415	3744.7452
15		47	$1\frac{1}{2}$	176.7150	1321.5454	25	6	80	$1\frac{1}{4}$	510.7063	3819.2657
15	3	47	$10\frac{7}{8}$	182.6545	1365.9634	25	9	80	$10\frac{3}{4}$	520.7692	3894.5203
15	6	48	$8\frac{1}{4}$	188.6923	1407.5165	26		81	$8\frac{1}{8}$	530.9304	3970.5098
15	9	49	$5\frac{3}{4}$	194.8282	1457.0032	26	3	82	$5\frac{1}{4}$	541.1896	4047.2322
16		50	$3\frac{1}{8}$	201.0624	1503.6250	26	6	83	3	551.5471	4124.6898
16	3	51	$0\frac{1}{2}$	207.3946	1550.9797	26	9	84	$0\frac{3}{8}$	562.0027	4202.9610
16	6	51	10	213.8251	1599.0696	27		84	$9\frac{7}{8}$	572.5566	4281.8072
16	9	52	$7\frac{3}{8}$	220.3537	1647.8930	27	3	85	$8\frac{1}{8}$	583.2085	4361.4664
17		53	$4\frac{7}{8}$	226.9806	1697.4516	27	6	86	$4\frac{5}{8}$	593.9587	4441.8607
17	3	54	$2\frac{1}{8}$	233.7055	1747.7431	27	9	87	$2\frac{1}{8}$	604.8070	4522.9886
17	6	54	$11\frac{5}{8}$	240.5287	1798.7698	28		87	$11\frac{1}{2}$	615.7536	4604.8517
17	9	55	$9\frac{1}{8}$	247.4500	1850.5301	28	3	88	9	626.7982	4686.4876
18		56	$6\frac{1}{2}$	254.4696	1903.0254	28	6	89	$6\frac{3}{8}$	637.9411	4770.7787
18	3	57	4	261.5872	1956.2537	28	9	90	$3\frac{3}{4}$	649.1821	4854.8434
18	6	58	$1\frac{3}{8}$	268.8031	2010.2171	29		91	$1\frac{1}{4}$	660.5214	4939.6432
18	9	58	$10\frac{3}{4}$	276.1171	2064.9140	29	3	91	$10\frac{5}{8}$	671.9587	5025.1759
19		59	$8\frac{1}{4}$	283.5294	2120.3462	29	6	92	$8\frac{1}{8}$	683.4943	5111.4487
19	3	60	$5\frac{5}{8}$	291.0397	2176.5113	29	9	93	$5\frac{1}{2}$	695.1280	5198.4451
19	6	61	$3\frac{1}{8}$	298.6483	2233.2914	30		94	$2\frac{7}{8}$	706.8600	5286.1818
19	9	62	$0\frac{1}{2}$	306.3550	2291.0452	30	3	95	$0\frac{3}{8}$	718.6900	5374.6512
20		62	$9\frac{7}{8}$	314.1600	2349.4141	30	6	95	$9\frac{3}{4}$	730.6183	5463.8558
20	3	63	$\frac{3}{8}$	322.0630	2408.5159	30	9	96	$7\frac{1}{4}$	742.6447	5553.7940

LOGARITHMS.

The logarithm of a number is the exponent of a power to which another given invariable number must be raised in order to produce the first number. Thus, in the common system of logarithms, in which the invariable number is 10, the logarithm of 1000 is 3, because 10 raised to the third power is 1000. In general, if $a^x = y$, in which equation a is a given invariable number, then x is the logarithm of y. All absolute numbers whether positive or negative, whole or fractional, may be produced by raising an invariable number to suitable powers. The invariable number is called the base of the system of logarithms; it may be any number whatever greater or less than unity; but having been once chosen, it must remain the same for the formation of all numbers in the same system. Whatever number may be selected for the base, the logarithm of the base is 1, and the logarithm of 1 is 0.

These properties of logarithms are of very great importance in facilitating the arithmetical operations of multiplication and division. For, if a multiplication is to be effected, it is only necessary to take from the logarithmic tables the logarithms of the factors, and add them into one sum, which gives the logarithm of the required product; and, on finding in the table the number corresponding to this new logarithm, the product itself is obtained. Thus, by means of a table of logarithms, the operation of multiplication is performed by simple addition. In like manner, if one number is to be divided by another, it is only necessary to subtract the logarithm of the divisor from that of the dividend, and to find in the table the number corresponding to this difference, which number is the quotient required. Thus, the quotient of a division is obtained by simple subtraction.

LOGARITHMS OF NUMBERS FROM 0 TO 1000.*

No.	0	1	2	3	4	5	6	7	8	9	Prop.
0	0	00000	30103	47712	60206	69897	77815	84510	90309	95424	
10	00000	00432	00860	01283	01703	02118	02530	02938	03342	03742	415
11	04139	04532	04921	05307	05690	06069	06445	06818	07188	07554	379
12	07918	08278	08636	08990	09342	09691	10037	10380	10721	11059	349
13	11394	11727	12057	12385	12710	13033	13353	13672	13987	14301	323
14	14613	14921	15228	15533	15836	16136	16435	16731	17026	17318	300
15	17609	17897	18184	18469	18752	19033	19312	19590	19865	20139	281
16	20412	20682	20951	21218	21484	21748	22010	22271	22530	22788	264
17	23045	23299	23552	23804	24054	24303	24551	24797	25042	25285	249
18	25527	25767	26007	26245	26481	26717	26951	27184	27415	27646	236
19	27875	28103	28330	28555	28780	29003	29225	29446	29666	29885	223
20	30103	30319	30535	30749	30963	31175	31386	31597	31806	32014	212
21	32222	32428	32633	32838	33041	33243	33445	33646	33845	34044	202
22	34242	34439	34635	34830	35024	35218	35410	35602	35793	35983	194
23	36173	36361	36548	36735	36921	37106	37291	37474	37657	37839	185
24	38021	38201	38381	38560	38739	38916	39093	39269	39445	39619	177
25	39794	39967	40140	40312	40483	40654	40824	40993	41162	41330	171
26	41497	41664	41830	41995	42160	42324	42488	42651	42813	42975	164
27	43436	43296	43456	43616	43775	43933	44090	44248	44404	44560	158
28	44716	44870	45024	45178	45331	45484	45636	45788	45939	46089	153
29	46240	46389	46538	46686	46834	46982	47129	47275	47421	47567	148
30	47712	47856	48000	48144	48287	48430	48572	48713	48855	48995	143
31	49136	49276	49415	49554	49693	49831	49968	50105	50242	50379	138
32	50515	50650	50785	50920	51054	51188	51321	51454	51587	51719	134
33	51851	51982	52113	52244	52374	52504	52633	52763	52891	53020	130
34	53148	53275	53102	53529	53655	53781	53907	54033	54157	54282	126
35	54407	54530	54654	54777	54900	55022	55145	55266	55388	55509	122
36	55630	55750	55870	55990	56410	56229	56348	56466	56584	56702	119
37	56820	56937	57054	57170	57287	57403	57518	57634	57749	57863	116
38	57978	58002	58206	58319	58433	58546	58658	58771	58883	58995	113
39	59106	59217	59328	59439	59549	59659	59769	59879	59988	60097	110
40	60206	60314	60422	60530	60638	60745	60852	60959	61066	61172	107
41	61278	61384	61489	61595	61700	61804	61909	62013	62117	62221	104
42	62325	62428	62531	62634	62736	62838	62941	63042	63144	63245	102
43	63347	63447	63548	63648	63749	63848	63948	64048	64147	64246	99
44	64345	64443	64542	64640	64738	64836	64933	65030	65127	65224	98
45	65321	65417	65513	65609	65075	65801	65896	65991	66086	66181	96
46	66276	66370	66464	66558	66651	66745	66838	66931	67024	67117	94
47	67210	67302	67394	67486	67577	67669	67760	67851	67942	68033	92
48	68124	68214	68304	68394	68484	68574	68663	68752	68842	68930	90
49	69020	69108	69196	69284	69372	69460	69548	69635	69722	69810	88
50	69897	69983	70070	70156	70243	70329	70415	70500	70586	70671	86
51	70757	70842	70927	71011	71096	71180	71265	71349	71433	71516	84
52	71600	71683	71767	71850	71933	72015	72098	72181	72263	72345	82
53	72428	72509	72591	72672	72754	72835	72916	72997	73078	73158	81
54	73239	73319	73399	73480	73559	73639	73719	73798	73878	73957	80
55	74036	74115	74193	74272	74351	74429	74507	74585	74663	74741	78
56	74818	74896	74973	75050	75127	75204	75281	75358	75434	75511	77
57	75587	75663	75739	75815	75891	75966	76042	76117	76192	76267	75
58	76342	76417	76492	76566	76641	76715	76789	76863	76937	77011	74
59	77085	77158	77232	77305	77378	77451	77524	77597	77670	77742	73
60	77815	77887	77959	78031	78103	78175	78247	78318	78390	78461	72
61	78533	78604	78675	78746	78816	78887	78958	79028	79098	79169	71
62	79239	79309	79379	79448	79518	79588	79657	79726	79796	79865	70
63	79934	80002	80071	80140	80208	80277	80345	80413	80482	80550	69

* Each logarithm is supposed to have the decimal sign . before it.

LOGARITHMS OF NUMBERS FROM 0 TO 1000.

(*Continued.*)

No.	0	1	2	3	4	5	6	7	8	9	Prop.
64	80618	80685	80753	80821	80888	80956	81023	81090	81157	81224	68
65	81291	81358	81424	81491	81557	81624	81690	81756	81822	81888	67
66	81954	82020	82085	82151	82216	82282	82347	82412	82477	82542	66
67	82607	82672	82736	82801	82866	82930	82994	83058	83123	83187	65
68	83250	83314	83378	83442	83505	83569	83632	83695	83758	83821	64
69	83884	83947	84010	84073	84136	84198	84260	84323	84385	84447	63
70	84509	84571	84633	84695	84757	84818	84880	84941	85003	85064	62
71	85125	85187	85248	85309	85369	85430	85491	85551	85612	85672	61
72	85733	85793	85853	85913	85973	86033	86093	86153	86213	86272	60
73	86332	86391	86451	86510	86569	86628	86687	86746	86805	86864	59
74	86923	86981	87040	87098	87157	87215	87273	87332	87390	87448	58
75	87506	87564	87621	87679	87737	87794	87852	87909	87966	88024	57
76	88081	88138	88195	88252	88309	88366	88422	88479	88536	88592	56
77	88649	88705	88761	88818	88874	88930	88986	89042	89098	89153	56
78	89209	89265	89320	89376	89431	89487	89542	89597	89652	89707	55
79	89762	89817	89872	89927	89982	90036	90091	90145	90200	90254	54
80	90309	90363	90417	90471	90525	90579	90633	90687	90741	90794	54
81	90848	90902	90955	91009	91062	91115	91169	91222	91275	91328	53
82	91381	91434	91487	91540	91592	91645	91698	91750	91803	91855	53
83	91907	91960	92012	92064	92116	92168	92220	92272	92324	92376	52
84	92427	92479	92531	92582	92634	92685	92737	92788	92839	92890	51
85	92941	92993	93044	93095	93146	93196	93247	93298	93348	93399	51
86	93449	93500	93550	93601	93651	93701	93751	93802	93852	93902	50
87	93951	94001	94051	94101	94151	94200	94250	94300	94349	94398	49
88	94448	94497	94546	94596	94645	94694	94743	94792	94841	94890	49
89	94939	94987	95036	95085	95133	95182	95230	95279	95327	95376	48
90	95424	95472	95520	95568	95616	95664	95712	95760	95808	95856	48
91	95904	95951	95999	96047	96094	96142	96189	96236	96284	96331	48
92	96378	96426	96473	96520	96507	96614	96661	96708	96754	96801	47
93	96848	96895	96941	96988	97034	97081	97127	97174	97220	97266	47
94	97312	97359	97405	97451	97497	97543	97589	97635	97680	97726	46
95	97772	97818	97863	97909	97954	98000	98045	98091	98136	98181	46
96	98227	98272	98317	98362	98407	98452	98497	98542	98587	98632	45
97	98677	98721	98766	98811	98855	98900	98945	98989	99033	99078	45
98	99122	99166	99211	99255	99299	99343	99387	99431	99475	99519	44
99	99563	99607	99657	99694	99738	99782	99825	99869	99913	99956	44

HYPERBOLIC LOGARITHMS.

Hyperbolic logarithms is a system of logarithms so called because the numbers express the areas between the asymptote and curve of the hyperbola. The hyperbolic logarithm of any number is to the common logarithm of the same number in the ratio of 2·30258509 to 1, or as 1 to ·43429448.

TABLE

OF HYPERBOLIC LOGARITHMS.

Num.	Log.	Num.	Log.	Num.	Log.	Num.	Log.
1.01	.0099	1.43	.3576	1.85	.6151	2.27	.8197
1.02	.0198	1.44	.3646	1.86	.6205	2.28	.8241
1.03	.0295	1.45	.3715	1.87	.6259	2.29	.8285
1.04	.0392	1.46	.3784	1.88	.6312	2.30	.8329
1.05	.0487	1.47	.3852	1.89	.6365	2.31	.8372
1.06	.0582	1.48	.3920	1.90	.6418	2.32	.8415
1.07	.0676	1.49	.3987	1.91	.6471	2.33	.8458
1.08	.0769	1.50	.4054	1.92	.6523	2.34	.8501
1.09	.0861	1.51	.4121	1.93	.6575	2.35	.8544
1.10	.0953	1.52	.4187	1.94	.6626	2.36	.8586
1.11	.1043	1.53	.4252	1.95	.6678	2.37	.8628
1.12	.1133	1.54	.4317	1.96	.6729	2.38	.8671
1.13	.1222	1.55	.4382	1.97	.6780	2.39	.8712
1.14	.1310	1.56	.4446	1.98	.6830	2.40	.8754
1.15	.1397	1 57	.4510	1.99	.6881	2.41	.8796
1.16	.1484	1.58	.4574	2.00	.6931	2.42	.8837
1.17	.1570	1.59	.4637	2.01	.6981	2.43	.8878
1.18	.1655	1.60	.4700	2.02	.7030	2.44	.8919
1.19	.1739	1 61	.4762	2.03	.7080	2.45	.8960
1.20	.1823	1.62	.4824	2.04	.7129	2.46	.9001
1.21	.1962	1.63	.4885	2.05	.7178	2.47	.9042
1.22	.1988	1.64	.4946	2.06	.7227	2.48	.9082
1.23	.2070	1.65	.5007	2.07	.7275	2.49	.9122
1.24	.2151	1.66	.5068	2.08	.7323	2.50	.9162
1.25	.2231	1.67	.5128	2.09	.7371	2.51	.9202
1.26	.2341	1.68	.5187	2.10	.7419	2.52	.9242
1.27	.2390	1.69	.5247	2.11	.7466	2.53	.9282
1.28	.2468	1.70	.5306	2.12	.7514	2.54	.9321
1.29	.2546	1.71	.5364	2.13	.7561	2.55	.9360
1.30	.2623	1.72	.5423	2.14	.7608	2.56	.9400
1.31	.2700	1.73	.5481	2.15	.7654	2.57	.9439
1.32	.2776	1.74	.5538	2.16	.7701	2.58	.9477
1.33	.2851	1.75	.5596	2.17	.7747	2.59	.9516
1.34	.2926	1.76	.5653	2.18	.7793	2.60	.9555
1.35	.3001	1.77	.5709	2.19	.7839	2.61	.9593
1.36	.3074	1.78	.5766	2.20	.7884	2.62	.9631
1.37	.3148	1.79	.5822	2.21	.7929	2.63	.9669
1.38	.3220	1.80	.5877	2.22	.7975	2.64	.9707
1.39	.3293	1.81	.5933	2.23	.8021	2.65	.9745
1.40	.3364	1.82	.5988	2.24	.8064	2.66	.9783
1.41	.3435	1.83	.6043	2.25	.8109	2.67	.9820
1.42	.3506	1.84	.6097	2.26	.8153	2.68	.9858

TABLE—(*Continued*)

OF HYPERBOLIC LOGARITHMS.

Num.	Log.	Num.	Log.	Num.	Log.	Num.	Log.
2.69	.9895	3.11	1.1346	3.53	1.2612	3.95	1.3737
2.70	.9932	3.12	1.1378	3.54	1.2641	3.96	1.3726
2.71	.9969	3.13	1.1410	3.55	1.2669	3.97	1.3787
2.72	1.0006	3.14	1.1442	3 56	1.2697	3.98	1.3812
2.73	1.0043	3.15	1.1474	3.57	1.2725	3.99	1.3837
2.74	1.0079	3.16	1.1505	3.58	1.2753	4.00	1.3862
2.75	1.0116	3.17	1.1537	3.59	1.2781	4.01	1.3887
2.76	1 0152	3.18	1.1568	3.60	1.2809	4.02	1.3912
2.77	1.0188	3.19	1.1600	3.61	1.2837	4.03	1.3937
2 78	1.0224	3.20	1.1631	3.62	1.2864	4.04	1.3962
2.79	1.0260	3.21	1.1662	3.63	1.2892	4.05	1.3987
2.80	1.0296	3.22	1.1693	3.64	1.2919	4.06	1.4011
2.81	1.0331	3.23	1.1724	3.65	1.2947	4.07	1.4036
2.82	1.0367	3.24	1.1755	3.66	1.2974	4.08	1.4060
2.83	1.0402	3.25	1.1786	3 67	1.3001	4.09	1·4085
2.84	1.0438	3.26	1.1817	3.68	1.3029	4.10	1.4109
2.85	1.0473	3.27	1.1847	3.69	1.3056	4.11	1.4134
2.86	1.0508	3.28	1.1878	3.70	1.3083	4.12	1.4158
2.87	1.0543	3.29	1.1908	3.71	1.3110	4.13	1.4182
2 88	1.0577	3.30	1.1939	3.72	1.3137	4.14	1.4206
2.89	1.0612	3.31	1.1969	3.73	1.3164	4.15	1.4231
2.90	1.0647	3.32	1.1999	3.74	1.3190	4.16	1.4255
2.91	1.0681	3.33	1 2029	3.75	1.3217	4.17	1.4279
2.92	1.0715	3.34	1.2059	3.76	1.3244	4.18	1.4303
2.93	1.0750	3.35	1.2089	3.77	1.3271	4.19	1.4327
2.94	1.0784	3.36	1.2119	3.78	1.3297	4.20	1.4350
2.95	1.0818	3.37	1.2149	3.79	1.3323	4.21	1.4374
2.96	1.0851	3.38	1.2178	3.80	1.3350	4.22	1.4398
2.97	1.0885	3.39	1.2208	3.81	1.3376	4.23	1.4421
2.98	1 0919	3.40	1.2237	3.82	1.3402	4.24	1.4445
2.99	1.0952	3.41	1.2267	3.83	1.3428	4.25	1.4469
3.00	1.0986	3.42	1.2296	3.84	1.3454	4.26	1.4492
3 01	1.1019	3.43	1.2325	3.85	1.3480	4.27	1.4516
3 02	1.1052	3.44	1.2354	3.86	1.3506	4.28	1.4539
3.03	1.1085	3.45	1.2387	3.87	1.3532	4.29	1.4562
3.04	1.1118	3.46	1.2412	3.88	1.3558	4.30	1.4586
3.05	1.1151	3.47	1.2441	3.89	1.3584	4.31	1.4609
3.06	1.1184	3.48	1.2470	3.90	1.3609	4.32	1.4632
3.07	1.1216	3.49	1.2499	3.91	1.3635	4.33	1.4655
3.08	1.1249	3.50	1.2527	3.92	1.3660	4.34	1.4678
3.09	1.1281	3.51	1.2556	3.93	1.3686	4.35	1.4701
3.10	1.1314	3.52	1.2584	3.94	1.3711	4.36	1.4724

TABLE—(*Concluded*)
OF HYPERBOLIC LOGARITHMS.

Num.	Log.	Num.	Log.	Num.	Log.	Num.	Log.
4.37	1.4747	4.79	1.5665	5.21	1.6505	5.63	1.7281
4.38	1.4778	4.80	1.5686	5.22	1.6524	5.64	1.7298
4.39	1.4793	4.81	1.5706	5.23	1.6544	5.65	1.7316
4.40	1.4816	4.82	1.5727	5.24	1.6563	5.66	1.7334
4.41	1.4838	4.83	1.5748	5.25	1.6582	5.67	1.7351
4.42	1.4838	4.84	1.5769	5.26	1.6601	5.68	1.7369
4.43	1.4883	4.85	1.5789	5.27	1.6620	5.69	1.7387
4.44	1.4906	4.86	1.5810	5.28	1.6639	5.70	1.7404
4.45	1.4929	4.87	1.5830	5.29	1.6658	5.71	1.7422
4.46	1.4914	4.88	1.5851	5.30	1.6677	5.72	1.7439
4.47	1.4973	4.89	1.5870	5.31	1.6695	5.73	1.7457
4.48	1.4996	4.90	1.5892	5.32	1.6714	5.74	1.7474
4.49	1.5018	4.91	1.5912	5.33	1.6733	5.75	1.7491
4.50	1.5040	4.92	1.5933	5.34	1.6752	5.76	1.7509
4.51	1.5062	4.93	1.5953	5.35	1.6770	5.77	1.7526
4.52	1.5085	4.94	1.5973	5.36	1.6789	5.78	1.7544
4.53	1.5107	4.95	1.5993	5.37	1.6808	5.79	1.7561
4.54	1.5129	4.96	1.6014	5.38	1.6826	5.80	1.7578
4.55	1.5151	4.97	1.6034	5.39	1.6845	5.81	1.7595
4.56	1.5173	4.98	1.6054	5.40	1.6863	5.82	1.7613
4.57	1.5195	4.99	1.6074	5.41	1.6882	5.83	1.7630
4.58	1.5216	5.00	1.6094	5.42	1.6900	5.84	1.7647
4.59	1.5238	5.01	1.6114	5.43	1.6919	5.85	1.7664
4.60	1.5260	5.02	1.6134	5.44	1.6937	5.86	1.7681
4.61	1.5282	5.03	1.6154	5.45	1.6956	5.87	1.7698
4.62	1.5303	5.04	1.6174	5.46	1.6974	5.88	1.7715
4.63	1.5325	5.05	1.6193	5.47	1.6992	5.89	1.7732
4.64	1.5347	5.06	1.6213	5.48	1.7011	5.90	1.7749
4.65	1.5368	5.07	1.6233	5.49	1.7029	5.91	1.7766
4.66	1.5390	5.08	1.6253	5.50	1.7047	5.92	1.7783
4.67	1.5411	5.09	1.6272	5.51	1.7065	5.93	1.7800
4.68	1.5432	5.10	1.6292	5.52	1.7083	5.94	1.7817
4.69	1.5454	5.11	1.6311	5.53	1.7101	5.95	1.7833
4.70	1.5475	5.12	1.6331	5.54	1.7119	5.96	1.7850
4.71	1.5496	5.13	1.6351	5.55	1.7137	5.97	1.7867
4.72	1.5518	5.14	1.6370	5.56	1.7155	5.98	1.7884
4.73	1.5539	5.15	1.6389	5.57	1.7173	5.99	1.7900
4.74	1.5560	5.16	1.6409	5.58	1.7191	6.00	1.7917
4.75	1.5581	5.17	1.6428	5.59	1.7209	6.01	1.7934
4.76	1.5602	5.18	1.6448	5.60	1.7227	6.02	1.7950
4.77	1.5623	5.19	1.6463	5.61	1.7245	6.03	1.7967
4.78	1.5644	5.20	1.6486	5.62	1.7263	6.04	1.7989

TABLE

CONTAINING THE DIAMETERS, CIRCUMFERENCES, AND AREAS OF CIRCLES FROM $\frac{1}{16}$ OF AN INCH TO 100 INCHES, ADVANCING BY $\frac{1}{16}$ OF AN INCH UP TO 10 INCHES, AND BY $\frac{1}{8}$ OF AN INCH FROM 10 TO 100 INCHES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
$\frac{1}{16}$	.1963	.0030	$\frac{7}{16}$	7.6576	4.6664
$\frac{1}{8}$	.3927	.0122	$\frac{1}{2}$	7.8540	4.9087
$\frac{3}{16}$	.5890	.0276	$\frac{9}{16}$	8.0503	5.1573
$\frac{1}{4}$	.7854	.0490	$\frac{5}{8}$	8.2467	5.4119
$\frac{5}{16}$	.9817	.0767	$\frac{11}{16}$	8.4430	5.6727
$\frac{3}{8}$	1.1781	.1104	$\frac{3}{4}$	8.6394	5.9395
$\frac{7}{16}$	1.3744	.1503	$\frac{13}{16}$	8.8357	6.2126
$\frac{1}{2}$	1.5708	.1963	$\frac{7}{8}$	9.0321	6.4918
$\frac{9}{16}$	1.7671	.2485	$\frac{15}{16}$	9.2284	6.7772
$\frac{5}{8}$	1.9635	.3068	3	9.4248	7.0686
$\frac{11}{16}$	2.1598	.3712	$\frac{1}{16}$	9.6211	7.3662
$\frac{3}{4}$	2.3562	.4417	$\frac{1}{8}$	9.8175	7.6699
$\frac{13}{16}$	2.5525	.5185	$\frac{3}{16}$	10.0138	7.9798
$\frac{7}{8}$	2.7489	.6013	$\frac{1}{4}$	10.2120	8.2957
$\frac{15}{16}$	2.9452	.6903	$\frac{5}{16}$	10.4065	8.6179
1	3.1416	.7854	$\frac{3}{8}$	10.6029	8.9462
$\frac{1}{16}$	3.3379	.8861	$\frac{7}{16}$	10.7992	9.2806
$\frac{1}{8}$	3.5343	.9940	$\frac{1}{2}$	10.9956	9.6211
$\frac{3}{16}$	3.7306	1.1075	$\frac{9}{16}$	11.1919	9.9678
$\frac{1}{4}$	3.9270	1.2271	$\frac{5}{8}$	11.3883	10.3206
$\frac{5}{16}$	4.1233	1.3529	$\frac{11}{16}$	11.5846	10.6796
$\frac{3}{8}$	4.3197	1.4848	$\frac{3}{4}$	11.7810	11.0446
$\frac{7}{16}$	4.5160	1.6229	$\frac{13}{16}$	11.9773	11.4159
$\frac{1}{2}$	4.7124	1.7671	$\frac{7}{8}$	12.1737	11.7932
$\frac{9}{16}$	4.9087	1.9175	$\frac{15}{16}$	12.3700	12.1768
$\frac{5}{8}$	5.1051	2.0739	4	12.5664	12.5664
$\frac{11}{16}$	5.3014	2.2365	$\frac{1}{16}$	12.7627	12.9622
$\frac{3}{4}$	5.4978	2.4052	$\frac{1}{8}$	12.9591	13.3640
$\frac{13}{16}$	5.6941	2.5801	$\frac{3}{16}$	13.1554	13.7721
$\frac{7}{8}$	5.8905	2.7611	$\frac{1}{4}$	13.3518	14.1862
$\frac{15}{16}$	6.0868	2.9483	$\frac{5}{16}$	13.5481	14.6066
2	6.2832	3.1416	$\frac{3}{8}$	13.7445	15.0331
$\frac{1}{16}$	6.4795	3.3411	$\frac{7}{16}$	13.9408	15.4657
$\frac{1}{8}$	6.6759	3.5465	$\frac{1}{2}$	14.1372	15.9043
$\frac{3}{16}$	6.8722	3.7582	$\frac{9}{16}$	14.3335	16.3492
$\frac{1}{4}$	7.0686	3.9760	$\frac{5}{8}$	14.5299	16.8001
$\frac{5}{16}$	7.2640	4.2001	$\frac{11}{16}$	14.7262	17.2573
$\frac{3}{8}$	7.4613	4.4302	$\frac{3}{4}$	14.9226	17.7205

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

Diam.	Circum.	Area.	Diam.	Circum.	Area.
Inch.			Inch.		
$\frac{13}{16}$	15.1189	18.1900	$\frac{7}{16}$	23.3656	43.4455
$\frac{7}{8}$	15.3153	18.6655	$\frac{1}{2}$	23.5620	44.1787
$\frac{15}{16}$	15.5716	19.1472	$\frac{9}{16}$	23.7583	44.9181
5	15.7080	19.6350	$\frac{5}{8}$	23.9547	45.6636
$\frac{1}{16}$	15.9043	20.1290	$\frac{11}{16}$	24.1510	46.4153
$\frac{1}{8}$	16.1007	20.6290	$\frac{3}{4}$	24.3474	47.1730
$\frac{3}{16}$	16.2970	21.1252	$\frac{13}{16}$	24.5437	47.9370
$\frac{1}{4}$	16.4934	21.6475	$\frac{7}{8}$	24.7401	48.7070
$\frac{5}{16}$	16.6897	22.1661	$\frac{15}{16}$	24.9364	49.4833
$\frac{3}{8}$	16.8861	22.6907	8	25.1328	50.2656
$\frac{7}{16}$	17.0824	23.2215	$\frac{1}{16}$	25.3291	51.0541
$\frac{1}{2}$	17.2788	23.7583	$\frac{1}{8}$	25.5255	51.8486
$\frac{9}{16}$	17.4751	24.3014	$\frac{3}{16}$	25.7218	52.8994
$\frac{5}{8}$	17.6715	24.8505	$\frac{1}{4}$	25.9182	53.4562
$\frac{11}{16}$	17.8678	25.4058	$\frac{5}{16}$	26.1145	54.2748
$\frac{3}{4}$	18.0642	25.9672	$\frac{3}{8}$	26.3109	55.0885
$\frac{13}{16}$	18.2605	26.5348	$\frac{7}{16}$	26.5072	55.9138
$\frac{7}{8}$	18.4569	27.1085	$\frac{1}{2}$	26.7036	56.7451
$\frac{15}{16}$	18.6532	27.6884	$\frac{9}{16}$	26.8999	57.5887
6	18.8496	28.2744	$\frac{5}{8}$	27.0963	58.4264
$\frac{1}{16}$	19.0459	28.8665	$\frac{11}{16}$	27.2926	59.7762
$\frac{1}{8}$	19.2423	29.4647	$\frac{3}{4}$	27.4890	60.1321
$\frac{3}{16}$	19.4386	30.0798	$\frac{13}{16}$	27.6853	60.9943
$\frac{1}{4}$	19.6350	30.6796	$\frac{7}{8}$	27.8817	61.8625
$\frac{5}{16}$	19.8313	31.2964	$\frac{15}{16}$	28.0780	62.7369
$\frac{3}{8}$	20.0277	31.9192	9	28.2744	63.6174
$\frac{7}{16}$	20.2240	32.5481	$\frac{1}{16}$	28.4707	64.5041
$\frac{1}{2}$	20.4204	33.1831	$\frac{1}{8}$	28.6671	65.3968
$\frac{9}{16}$	20.6167	33.8244	$\frac{3}{16}$	28.8634	66.2957
$\frac{5}{8}$	20.8131	34.4717	$\frac{1}{4}$	29.0598	67.2007
$\frac{11}{16}$	21.0094	35.1252	$\frac{5}{16}$	29.2561	68.1120
$\frac{3}{4}$	21.2058	35.7847	$\frac{3}{8}$	29.4525	69.0293
$\frac{13}{16}$	21.4021	36.4505	$\frac{7}{16}$	29.6488	69.9528
$\frac{7}{8}$	21.5985	37.1224	$\frac{1}{2}$	29.8452	70.8823
$\frac{15}{16}$	21.7948	37.8005	$\frac{9}{16}$	30.0415	71.8181
7	21.9912	38.4846	$\frac{5}{8}$	30.2379	72.7599
$\frac{1}{16}$	22.1875	39.1749	$\frac{11}{16}$	30.4342	73.7079
$\frac{1}{8}$	22.3839	39.8713	$\frac{3}{4}$	30.6306	74.6620
$\frac{3}{16}$	22.5802	40.5469	$\frac{13}{16}$	30.8269	75.6223
$\frac{1}{4}$	22.7766	41.2825	$\frac{7}{8}$	31.0233	76.5887
$\frac{5}{16}$	22.9729	41.9974	$\frac{15}{16}$	31.2196	77.5613
$\frac{3}{8}$	23.1693	42.7184	10	31.4160	78.5400

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
1/8	31.8087	80.5157	3/8	48.3021	185.6612
1/4	32.2014	82.5160	1/2	48.6948	188.6923
3/8	32.5941	84.5409	5/8	49.0875	191.7480
1/2	32.9868	86.5903	3/4	49.4802	194.8282
5/8	33.3795	88.6643	7/8	49.8729	197.9330
3/4	33.7722	90.7627	16	50.2656	201.0624
7/8	34.1649	92.8858	1/8	50.6583	204.2162
11	34.5576	95.0334	1/4	51.0510	207.3946
1/8	34.9503	97.2053	3/8	51.4437	210.5976
1/4	35.3430	99.4021	1/2	51.8364	213.8251
3/8	35.7357	101.6234	5/8	52.2291	217.0772
1/2	36.1284	103.8691	3/4	52.6218	220.3537
5/8	36.5211	106.1394	7/8	53.0145	223.6549
3/4	36.9138	108.4342	17	53.4072	226.9806
7/8	37.3065	110.7536	1/8	53.7999	230.3308
12	37.6992	113.0976	1/4	54.1926	233.7055
1/8	38.0919	115.4660	3/8	54.5853	237.1049
1/4	38.4846	117.8590	1/2	54.9780	240.5287
3/8	38.8773	120.2766	5/8	55.3707	243.9771
1/2	39.2700	122.7187	3/4	55.7634	247.4500
5/8	39.6627	125.1854	7/8	56.1561	250.9475
3/4	40.0554	127.6765	18	56.5488	254.4696
7/8	40.4481	130.1923	1/8	56.9415	258.0161
13	40.8408	132.7326	1/4	57.3342	261.5872
1/8	41.2338	135.2974	3/8	57.7269	265.1829
1/4	41.6262	137.8867	1/2	58.1196	268.8031
3/8	42.0189	140.5007	5/8	58.5123	272.4479
1/2	42.4116	143.1391	3/4	58.9056	276.1171
5/8	42.8043	145.8021	7/8	59.2977	279.8110
3/4	43.1970	148.4896	19	59.6904	283.5294
7/8	43.5897	151.2017	1/8	60.0831	287.2723
14	43.9824	153.9384	1/4	60.4758	291.0397
1/8	44.3751	156.6995	3/8	60.8685	294.8312
1/4	44.7676	159.4852	1/2	61.2612	298.6483
3/8	45.1605	162.2956	5/8	61.6539	302.4894
1/2	45.5532	165.1303	3/4	62.0466	306.3550
5/8	45.9459	167.9896	7/8	62.4393	310.2452
3/4	46.3386	170.8735	20	62.8320	314.1600
7/8	46.7313	173.7820	1/8	63.2247	318.0992
15	47.1240	176.7150	1/4	63.6174	322.0630
1/8	47.5167	179.6725	3/8	64.0101	326.0514
1/4	47.9094	182.6545	1/2	64.4028	330.0643

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
5/8	64.7955	334.1018	7/8	81.2889	525.8375
3/4	65.1882	338.1637	26	81.6816	530.9304
7/8	65.5809	342.2503	1/8	82.0743	536.0477
21	65.7936	346.3614	1/4	82.4670	541.1896
1/8	66.3663	350.4970	3/8	82.8597	546.3561
1/4	66.7590	354.6571	1/2	83.2524	551.5471
3/8	67.1517	358.8419	5/8	83.6451	556.7627
1/2	67.5444	363.0511	3/4	84.0378	562.0027
5/8	67.9371	367.2849	7/8	84.4305	567.2674
3/4	68.3298	371.5432	27	84.8232	572.5566
7/8	68.7225	375.8261	1/8	85.2159	577.8703
22	69.1152	380.1336	1/4	85.6086	583.2085
1/8	69.5079	384.4655	3/8	86.0013	588.5714
1/4	69.9006	388.8220	1/2	86.3940	593.9587
3/8	70.2933	393.2031	5/8	86.7867	599.3706
1/2	70.6860	397.6087	3/4	87.1794	604.8070
5/8	71.0787	402.0388	7/8	87.5721	610.2680
3/4	71.4714	406.4935	28	87.9648	615.7536
7/8	71.8641	410.9728	1/8	88.3575	621.2636
23	72.2568	415.4766	1/4	88.7502	626.7982
1/8	72.6495	420.0049	3/8	89.1429	632.3574
1/4	73.0422	424.5577	1/2	89.5356	637.9411
3/8	73.4349	429.1352	5/8	89.9283	643.5494
1/2	73.8276	433.7371	3/4	90.3210	649.1821
5/8	74.2203	438.3636	7/8	90.7137	654.8395
3/4	74.6130	443.0146	29	91.1064	660.5214
7/8	75.0057	447.6992	1/8	91.4991	666.2278
24	75.3984	452.3904	1/4	91.8918	671.9587
1/8	75.7911	457.1150	3/8	92.2845	677.7143
1/4	76.1838	461.8642	1/2	92.6772	683.4943
3/8	76.5765	466.6380	5/8	93.0699	689.2989
1/2	76.9692	471.4363	3/4	93.4626	695.1280
5/8	77.3619	476.2592	7/8	93.8553	700.9817
3/4	77.7546	481.1065	30	94.2480	706.8600
7/8	78.1473	485.9785	1/8	94.6407	712.7627
25	78.5400	490.8750	1/4	95.0334	718.6900
1/8	78.9327	495.7960	3/8	95.4261	724.6419
1/4	79.3254	500.7415	1/2	95.8188	730.6183
3/8	79.7181	505.7117	5/8	96.2115	736.6193
1/2	80.1108	510.7063	3/4	96.6042	742.6447
5/8	80.5035	515.7255	7/8	96.9969	748.6948
3/4	80.8962	520.7692	31	97.3896	754.7694

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
$\frac{1}{8}$	97.7823	760.8685	$\frac{3}{8}$	114.2757	1039 1946
$\frac{1}{4}$	98.1750	766.9921	$\frac{1}{2}$	114.6684	1046.3941
$\frac{3}{8}$	98.5677	773.1404	$\frac{5}{8}$	115.0611	1053.5281
$\frac{1}{2}$	98.9684	779.3131	$\frac{3}{4}$	115.4538	1060.7317
$\frac{5}{8}$	99.3531	785.5104	$\frac{7}{8}$	115.8465	1067.9599
$\frac{3}{4}$	99.7458	791.7322	37	116.2392	1075.2126
$\frac{7}{8}$	100.1385	797.9786	$\frac{1}{8}$	116.6319	1082.4898
32	100.5312	804.2496	$\frac{1}{4}$	117.0246	1089.7915
$\frac{1}{8}$	100.9240	810.5450	$\frac{3}{8}$	117.4173	1097.1179
$\frac{1}{4}$	101.3166	816.8650	$\frac{1}{2}$	117.8100	1104.4687
$\frac{3}{8}$	101.7093	823.2096	$\frac{5}{8}$	118.2027	1111.8441
$\frac{1}{2}$	102.1020	829.5787	$\frac{3}{4}$	118.5954	1119.2440
$\frac{5}{8}$	102.4947	835.9724	$\frac{7}{8}$	118.9881	1126.6685
$\frac{3}{4}$	102.8874	842.3905	38	119.3808	1134.1176
$\frac{7}{8}$	103.2801	848.8333	$\frac{1}{8}$	119.7735	1141.5911
33	103.6728	855.3006	$\frac{1}{4}$	120.1662	1149.0892
$\frac{1}{8}$	104.0655	861.7924	$\frac{3}{8}$	120.5589	1156.6119
$\frac{1}{4}$	104.4582	868.3087	$\frac{1}{2}$	120.9516	1164.1591
$\frac{3}{8}$	104.8509	874.8497	$\frac{5}{8}$	121.3443	1171.7309
$\frac{1}{2}$	105.2436	881.4151	$\frac{3}{4}$	121.7370	1179.3271
$\frac{5}{8}$	105.6363	888.0051	$\frac{7}{8}$	122.1297	1186.9480
$\frac{3}{4}$	106.0290	894.6196	39	122.5224	1194.5934
$\frac{7}{8}$	106.4217	901.2587	$\frac{1}{8}$	122.9151	1202.2633
34	106.8144	907.9224	$\frac{1}{4}$	123.3078	1209.9577
$\frac{1}{8}$	107.2071	914.6105	$\frac{3}{8}$	123.7005	1217.6768
$\frac{1}{4}$	107.5998	921.3232	$\frac{1}{2}$	124.0932	1225.4203
$\frac{3}{8}$	107.9925	928.0605	$\frac{5}{8}$	124.4859	1233.1884
$\frac{1}{2}$	108.3852	934.8223	$\frac{3}{4}$	124.9787	1240.9810
$\frac{5}{8}$	108.7779	941.6086	$\frac{7}{8}$	125.2713	1248.7982
$\frac{3}{4}$	109.1706	948.4195	40	125.6640	1256.6400
$\frac{7}{8}$	109.5633	955.2550	$\frac{1}{8}$	126.0567	1264.5062
35	109.9560	962.1150	$\frac{1}{4}$	126.4494	1272.3970
$\frac{1}{8}$	110.3487	968.9995	$\frac{3}{8}$	126.8421	1280.3124
$\frac{1}{4}$	110.7414	975.9085	$\frac{1}{2}$	127.2348	1288.2523
$\frac{3}{8}$	111.1341	982.8422	$\frac{5}{8}$	127.6275	1296.2168
$\frac{1}{2}$	111.5268	989.8003	$\frac{3}{4}$	128.0202	1304.2057
$\frac{5}{8}$	111 9195	996.7830	$\frac{7}{8}$	128.4129	1312.2193
$\frac{3}{4}$	112.3122	1003.7902	41	128.8056	1320.2574
$\frac{7}{8}$	112.7049	1010.8220	$\frac{1}{8}$	129.1983	1328.3200
36	113.0976	1017.8784	$\frac{1}{4}$	129.5910	1336.4071
$\frac{1}{8}$	113.4903	1024.9592	$\frac{3}{8}$	129.9837	1344.5189
$\frac{1}{4}$	113.8830	1032.0646	$\frac{1}{2}$	130.3764	1352.6551

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
5/8	130.7691	1360.8159	7/8	147.2625	1725.7324
3/4	131.1618	1369.0012	47	147.6552	1734.9486
7/8	131.5545	1377.2111	1/8	148.0479	1744.1893
42	131.9472	1385.4456	1/4	148.4406	1753.4545
1/8	132.3399	1393.7045	3/8	148.8333	1762.7344
1/4	132.7326	1401.9880	1/2	149.2260	1772.0587
3/8	133.1253	1410.2961	5/8	149.6187	1781.3976
1/2	133.5180	1418.6287	3/4	150.0114	1790.7610
5/8	133.9107	1426.9859	7/8	150.4041	1800.1490
3/4	134.3034	1435.3675	48	150.7968	1809.5616
7/8	134.6961	1443.7738	1/8	151.1895	1818.9986
43	135.0888	1452.2046	1/4	151.5822	1828.4602
1/8	135.4815	1460.6599	3/8	151.9749	1837.9364
1/4	135.8742	1469.1397	1/2	152.3676	1847.4571
3/8	136.2669	1477.6342	5/8	152.7603	1856.9924
1/2	136.6596	1486.1731	3/4	153.1530	1868.5521
5/8	137.0523	1494.7266	7/8	153.5457	1876.1365
3/4	137.4450	1503.3046	49	153.9384	1885.7454
7/8	137.8377	1511.9072	1/8	154.3311	1895.3788
44	138.2304	1520.5344	1/4	154.7238	1905.0367
1/8	138.6231	1529.1860	3/8	155.1165	1914.7093
1/4	139.0158	1537.8622	1/2	155.5092	1924.4263
3/8	139.4085	1546.5530	5/8	155.9019	1934.1579
1/2	139.8012	1555.2883	3/4	156.2946	1943.9140
5/8	140.1939	1564.0382	7/8	156.6873	1953.6947
3/4	140.5866	1572.8125	50	157.0800	1963.5000
7/8	140.9793	1581.6115	1/8	157.4727	1973.3297
45	141.3720	1590.4350	1/4	157.8654	1983.1840
1/8	141.7647	1599.2830	3/8	158.2581	1993.0529
1/4	142.1574	1608.1555	1/2	158.6508	2002.9663
3/8	142.5501	1617.0427	5/8	159.0435	2012.8943
1/2	142.9428	1625.9743	3/4	159.4362	2022.8467
5/8	143.3355	1634.9205	7/8	159.8289	2032.8238
3/4	143.7382	1643.8912	51	160.2216	2042.8254
7/8	144.1209	1652.8865	1/8	160.6143	2052.8515
46	144.5136	1661.9064	1/4	161.0070	2062.9021
1/8	144.9063	1670.9507	3/8	161.3997	2072.9764
1/4	145.2990	1680.0196	1/2	161.7924	2083.0771
3/8	145.6917	1689.1031	5/8	162.1851	2093.2014
1/2	146.0844	1698.2311	3/4	162.5778	2103.3502
5/8	146.4771	1707.3737	7/8	162.9705	2113.5236
3/4	146.8698	1716.5407	52	163.3632	2123.7216

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
1/8	163.7559	2133.9440	3/8	180.2493	2585.4509
1/4	164.1486	2144.1910	1/2	180.6423	2596.7287
3/8	164.5413	2154.4626	5/8	181.0347	2608.0311
1/2	164.9340	2164.7587	3/4	181.4274	2619.3580
5/8	165.3267	2175.0794	7/8	181.8201	2630.7095
3/4	165.7194	2185.4245	58	182.2128	2642.0856
7/8	166.1121	2195.7943	1/8	182.6055	2653.4861
53	166.5048	2206.1886	1/4	182.9982	2664.9112
1/8	166.8975	2216.6074	3/8	183.3909	2676.3609
1/4	167.2902	2227.0507	1/2	183.7836	2687.8351
3/8	167.6829	2237.5187	5/8	184.1763	2699.3338
1/2	168.0756	2248.0111	3/4	184 5690	2710.8571
5/8	168.4683	2258.5281	7/8	184.9617	2722.4050
3/4	168.8610	2269.0696	59	185.3544	2733.9774
7/8	169.2537	2279.6357	1/8	185.7471	2745.5743
54	169.6464	2290.2264	1/4	186.1398	2757.1957
1/8	170.0391	2300.8415	3/8	186.5325	2768.8418
1/4	170.4318	2311.4812	1/2	186.9252	2780.5123
3/8	170.8245	2322.1455	5/8	187.3179	2792.2074
1/2	171.2172	2332.8343	3/4	187.7106	2803.9270
5/8	171.6099	2343.5477	7/8	188.1033	2815.6712
3/4	172.0026	2354.2855	60	188.4960	2827.4400
7/8	172.3593	2365.0480	1/8	188.8887	2839.2332
55	172.7880	2375.8350	1/4	189.2814	2851.0510
1/8	173.1807	2386.6465	3/8	189.6741	2862.8934
1/4	173.5734	2397.4825	1/2	190.0668	2874.7603
3/8	173.9661	2408.3432	5/8	190.4595	2886.6517
1/2	174.3588	2419.2283	3/4	190.8522	2898.5677
5/8	174.7515	2430.1833	7/8	191.2419	2910.5083
3/4	175.1442	2441.0772	61	191.6376	2922.4734
7/8	175.5369	2452.0310	1/8	192.0303	2934.4630
56	175.9296	2463.0144	1/4	192.4230	2946.4771
1/8	176.3323	2474.0222	3/8	192.8157	2958.5139
1/4	176.7150	2485.3546	1/2	193.2084	2970.5791
3/8	177.1077	2496.1116	5/8	193.6011	2982.6669
1/2	177.5004	2507.1931	3/4	193.9931	2994.7792
5/8	177.8931	2518.2992	7/8	194.3865	3006.9161
3/4	178.2858	2529.4297	62	194.7792	3019.0776
7/8	178.6785	2543.5849	1/8	195.1719	3031.2635
57	179.0712	2551.7646	1/4	195.5646	3043.4740
1/8	179.4639	2562.9688	3/8	195.9573	3055.7091
1/4	179.8566	2574.1975	1/2	196.3500	3067.9687

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
5/8	196.7427	3080.2529	7/8	213.2361	3618.3300
3/4	197.1354	3092.5615	68	213.6288	3631.6896
7/8	197.5281	3104.8948	1/8	214.0215	3645.0536
63	197.9208	3117.2526	1/4	214.4142	3658.4402
1/8	198.3135	3129.6349	3/8	214.8069	3671.8554
1/4	198.7062	3142.0417	1/2	215.1996	3685.2931
3/8	199.0989	3154.4732	5/8	215.5923	3698.7554
1/2	199.4916	3166.9291	3/4	215.9850	3712.2421
5/8	199.8843	3179.4096	7/8	216.3777	3725.7535
3/4	200.2770	3191.9146	69	216.7704	3739.2894
7/8	200.6697	3204.4442	1/8	217.1631	3752.8498
64	201.0624	3216.9984	1/4	217.5558	3766.4327
1/8	201.4551	3229.5770	3/8	217.9485	3780.0443
1/4	201.8478	3242.1782	1/2	218.3412	3793.6783
3/8	202.2405	3254.8080	5/8	218.7339	3807.3369
1/2	202.6332	3267.4603	3/4	219.1266	3821.0200
5/8	203.0259	3280.1372	7/8	219.5193	3834.7277
3/4	203.4186	3292.8385	70	219.9120	3848.4600
7/8	203.8113	3305.5645	1/8	220.3047	3862.2167
65	204.2040	3318.3151	1/4	220.6974	3875.9960
1/8	204.5917	3331.0900	3/8	221.0901	3889.8039
1/4	204.9894	3343.8875	1/2	221.4828	3903.6343
3/8	205.3821	3356.7137	5/8	221.8755	3917.4893
1/2	205.7748	3369.5623	3/4	222.2682	3931.3687
5/8	206.1675	3382.4355	7/8	222.6609	3945.2728
3/4	206.5602	3395.3332	71	223.0536	3959.2014
7/8	206.9529	3408.2555	1/8	223.4463	3973.1545
66	207.3456	3421.2024	1/4	223.8390	3987.1301
1/8	207.7383	3434.1737	3/8	224.2317	4001.1344
1/4	208.1310	3447.1676	1/2	224.6244	4015.1611
3/8	208.5237	3468.1901	5/8	225.0171	4029.2124
1/2	208.9164	3473.2351	3/4	225.4098	4043.2882
5/8	209.3091	3486.3047	7/8	225.8025	4057.3886
3/4	209.7018	3499.3987	72	226.1952	4071.5136
7/8	210.0945	3512.5174	1/8	226.5879	4085.6631
67	210.4872	3525.6606	1/4	226.9806	4099.8350
1/8	210.8799	3538.8283	3/8	227.3733	4114.0356
1/4	211.2726	3552.0185	1/2	227.7660	4128.2587
3/8	211.6653	3565.2374	5/8	228.1587	4142.5064
1/2	212.0580	3578.4787	3/4	228.5514	4156.7785
5/8	212.4507	3591.7446	7/8	228.9441	4171.0753
3/4	212.8434	3605.0350	73	229.3368	4185.3966

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
1/8	229.7295	4199.7424	3/8	246.2229	4824.4299
1/4	230.1222	4214.1107	1/2	246.6156	4839.8311
3/8	230.5149	4228.5077	5/8	247.0083	4855.2568
1/2	230.9076	4242.9271	3/4	247.4010	4870.7071
5/8	231.3003	4257.3711	7/8	247.7937	4886.1820
3/4	231.6930	4271.8396	79	248.1864	4901.6814
7/8	232.0857	4286.3327	1/8	248.5791	4917.2053
74	232.4784	4300.8504	1/4	248.9718	4932.7517
1/8	232.8711	4315.3926	3/8	249.3645	4948.3268
1/4	233.2638	4329.9572	1/2	249.7572	4963.9243.
3/8	233.6565	4344.5505	5/8	250.1499	4979.5456
1/2	234.0492	4359.1663	3/4	250.5426	4995.1930
5/8	234.4419	4373.8067	7/8	250.9353	5010.8642
3/4	234.8346	4388.4715	80	251.3280	5026.5600
7/8	235.2273	4403.1610	1/8	251.7207	5042.2803
75	235.6200	4417.8750	1/4	252.1134	5058.0230
1/8	236.0127	4432.6135	3/8	252.5061	5073.7944
1/4	236.4054	4447.3745	1/2	252.8988	5089.5883
3/8	236.7981	4462.1642	5/8	253.2915	5106.4060
1/2	237.1908	4476.9763	3/4	253.6842	5121.2497
5/8	237.5835	4491.8130	7/8	254.0769	5137.1173
3/4	237.9762	4506.6742	81	254.4696	5153.0094
7/8	238.3689	4521.5600	1/8	254.8623	5168.9260
76	238.7616	4536.4704	1/4	255.2550	5184.8651
1/8	239.1543	4551.4023	3/8	255.6477	5200.8329
1/4	239.5470	4566.3626	1/2	256.0404	5216.8231
3/8	239.9397	4581.3486	5/8	256.4331	5232.8371
1/2	240.3324	4596.3571	3/4	256.8258	5248.8772
5/8	240.7251	4611.3902	7/8	257.2105	5264.9411
3/4	241.1178	4626.4477	82	257.6112	5281.0296
7/8	241.5105	4641.3299	1/8	258.0039	5297.1426
77	241.9032	4656.6366	1/4	258.3966	5313.2780
1/8	242.2959	4671.7678	3/8	258.7893	5329.4421
1/4	242.6886	4686.9215	1/2	259.1820	5345.6287
3/8	243.0813	4702.1039	5/8	259.5747	5361.8391
1/2	243.4740	4717.3087	3/4	259.9674	5378.0755
5/8	243.8667	4732.5381	7/8	260.3601	5394.3358
3/4	244.2594	4747.7920	83	260.7528	5410.6206
7/8	244.6521	4763.0705	1/8	261.1455	5426.9299
78	245.0448	4778.3736	1/4	261.5382	5443.2617
1/8	245.4375	4793.7012	3/8	261.9309	5459.6222
1/4	245.8302	4809.0512	1/2	262.3236	5476.0051

TABLE—(*Continued*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
$\frac{5}{8}$	262.7163	5492.4118	$\frac{7}{8}$	279.2097	6203.6905
$\frac{3}{4}$	263.1090	5508.8446	89	279.6024	6221.1534
$\frac{7}{8}$	263.5017	5525.3012	$\frac{1}{8}$	279.9951	6238.6408
84	263.8944	5541.7824	$\frac{1}{4}$	280.3878	6256.1507
$\frac{1}{8}$	264.2871	5558.2881	$\frac{3}{8}$	280.7805	6273.6893
$\frac{1}{4}$	264.6798	5574.8162	$\frac{1}{2}$	281.1732	6291.2503
$\frac{3}{8}$	265.0725	5591.3730	$\frac{5}{8}$	281.5659	6308.8351
$\frac{1}{2}$	265.4652	5607.9523	$\frac{3}{4}$	281.9586	6326.4460
$\frac{5}{8}$	265.8579	5624.5554	$\frac{7}{8}$	282.3513	6344.0807
$\frac{3}{4}$	266.2506	5641.1845	90	282.7440	6361.7400
$\frac{7}{8}$	266.6433	5657.8357	$\frac{1}{8}$	283.1367	6379.4238
85	267.0360	5674.5150	$\frac{1}{4}$	283.5294	6397.1300
$\frac{1}{8}$	267.4287	5691.2170	$\frac{3}{8}$	283.9221	6414.8649
$\frac{1}{4}$	267.8214	5707.9415	$\frac{1}{2}$	284.3148	6432.6223
$\frac{3}{8}$	268.2141	5724.6947	$\frac{5}{8}$	284.7075	6450.4039
$\frac{1}{2}$	268.6068	5741.4703	$\frac{3}{4}$	285.1002	6468.2107
$\frac{5}{8}$	268.9997	5758.2697	$\frac{7}{8}$	285.4929	6486.0418
$\frac{3}{4}$	269.3922	5775.0952	91	285.8856	6503.8974
$\frac{7}{8}$	269.7849	5791.9445	$\frac{1}{8}$	286.2783	6521.7772
86	270.1776	5808.8184	$\frac{1}{4}$	286.6710	6539.6801
$\frac{1}{8}$	270.5703	5825.7168	$\frac{3}{8}$	287.0637	6557.6114
$\frac{1}{4}$	270.9630	5842.6376	$\frac{1}{2}$	287.4564	6573.5651
$\frac{3}{8}$	271.3557	5859.5871	$\frac{5}{8}$	287.8491	6593.5431
$\frac{1}{2}$	271.7484	5876.5591	$\frac{3}{4}$	288.2418	6611.5462
$\frac{5}{8}$	272.1411	5893.5549	$\frac{7}{8}$	288.6345	6629.5736
$\frac{3}{4}$	272.5338	5910.5767	92	289.0272	6647.6258
$\frac{7}{8}$	272.9265	5927.6224	$\frac{1}{8}$	289.4199	6665.7021
87	273.3192	5944.6926	$\frac{1}{4}$	289.8125	6683.8010
$\frac{1}{8}$	273.7119	5961.7873	$\frac{3}{8}$	290.2053	6701.9286
$\frac{1}{4}$	274.1046	5978.9045	$\frac{1}{2}$	290.5980	6720.0787
$\frac{3}{8}$	274.4973	5996.0504	$\frac{5}{8}$	290.9907	6738.2530
$\frac{1}{2}$	274.8900	6013.2187	$\frac{3}{4}$	291.3834	6756.4525
$\frac{5}{8}$	275.2827	6030.4108	$\frac{7}{8}$	291.7661	6774.6763
$\frac{3}{4}$	275.6754	6047.6290	93	292.1688	6792.9248
$\frac{7}{8}$	276.0681	6064.8710	$\frac{1}{8}$	292.5615	6811.1974
88	276.4608	6082.1376	$\frac{1}{4}$	292.9542	6829.4927
$\frac{1}{8}$	276.8535	6099.4287	$\frac{3}{8}$	293.3469	6847.8167
$\frac{1}{4}$	277.2462	6116.7422	$\frac{1}{2}$	293.7396	6866.1631
$\frac{3}{8}$	277.6389	6134.0844	$\frac{5}{8}$	294.1323	6884.5338
$\frac{1}{2}$	278.0316	6151.4491	$\frac{3}{4}$	294.5350	6902.9296
$\frac{5}{8}$	278.4243	6169.8376	$\frac{7}{8}$	294.9177	6921.3497
$\frac{3}{4}$	278.8170	6186.2591	94	295.3104	6939.7946

TABLE—(*Concluded*)

CONTAINING THE DIAM., CIRCUMFERENCES, AND AREAS OF CIRCLES.

DIAM.	CIRCUM.	AREA.	DIAM.	CIRCUM.	AREA.
Inch.			Inch.		
1/8	295.7031	6958.2636	1/8	305.1279	7408.8868
1/4	296.0958	6976.7552	1/4	305.5206	7427.9675
3/8	296.4885	6995.2755	3/8	305.9133	7447.0769
1/2	296.8812	7013.8183	1/2	306.3060	7466.2087
5/8	297.2739	7032.3853	5/8	306.6987	7485.3648
3/4	297.6666	7050.9775	3/4	307.0914	7504.5460
7/8	298.0593	7069.5940	7/8	307.4841	7523.7515
95	298.4520	7088.2352	98	307.8768	7542.9818
1/8	298.8447	7106.9005	1/8	308.2695	7562.2362
1/4	299.2374	7125.5885	1/4	308.6622	7581.5132
3/8	299.6301	7144.3052	3/8	309.0549	7600.8189
1/2	300.0228	7163.0443	1/2	309.4476	7620.1471
5/8	300.4155	7181.8077	5/8	309.8403	7639.4995
3/4	300.8082	7200.5962	3/4	310.2330	7658.8771
7/8	301.2009	7219.4090	7/8	310.6257	7678.2790
96	301.5936	7238.2466	99	311.0184	7697.7056
1/8	301.9863	7257.1083	1/8	311.4111	7717.1563
1/4	302.3790	7275.9926	1/4	311.8038	7736.6297
3/8	302.7717	7294.9056	3/8	312.1965	7756.1318
1/2	303.1644	7313.8411	1/2	312.5892	7775.6563
5/8	303.5571	7332.8008	5/8	312.9819	7795.2051
3/4	303.9498	7351.7857	3/4	313.3746	7814.7790
7/8	304.3425	7370.7949	7/8	313.7673	7834.3772
97	304.7352	7389.8288	100	314.1600	7854.0000

For the Circumference of Larger Circles than those contained in the Tables.—Multiply the diameter by 3·1416.

EXAMPLE.

Diam. in inches,	110
	3·1416
	345·5760

For areas larger than those in the Tables, see page 489.

TABLE

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS OF ALL NUMBERS FROM 1 TO 620.

Number.	Square.	Cube.	Square Root.	Cube Root.
1	1	1	1.	1.
2	4	8	1.4142 136	1.2599 21
3	9	27	1.7230 508	1.4422 496
4	16	64	2.	1.5874 011
5	25	125	2.2360 68	1.7099 759
6	36	216	2.4494 897	1.8171 206
7	49	343	2.6457 513	1.9129 312
8	64	512	2.8284 271	2.
9	81	729	3.	2.0800 837
10	1 00	1 000	3.1622 777	2.1544 347
11	1 21	1 331	3.3166 248	2.2239 801
12	1 44	1 728	3.4641 016	2.2894 286
13	1 69	2 197	3.6055 513	2.3513 347
14	1 96	2 744	3.7416 574	2.4101 422
15	2 25	3 375	3.8729 833	2.4662 121
16	2 56	4 096	4.	2.5198 421
17	2 89	4 913	4.1231 056	2.5712 816
18	3 24	5 832	4.2426 407	2.6207 414
19	3 61	6 859	4.3585 989	2.6684 016
20	4 00	8 000	4.4721 36	2.7144 177
21	4 41	9 261	4.5825 757	2.7589 243
22	4 84	10 648	4.6904 158	2.8020 393
23	5 29	12 167	4.7958 315	2.8438 67
24	5 76	13 824	4.8989 795	2.8844 991
25	6 25	15 625	5.	2.9240 177
26	6 76	17 576	5.0990 195	2.9224 96
27	7 29	19 683	5.1961 524	3.
28	7 84	21 952	5.2915 026	3.0365 889
29	8 41	24 389	5.3851 648	3.0723 168
30	9 00	27 000	5.4772 256	3.1072 325
31	9 61	29 791	5.5677 644	3.1413 806
32	10 24	32 768	5.6568 542	3.1748 021
33	10 89	35 937	5.7445 626	3.2075 343
34	11 56	39 304	5.8309 519	3.2396 118
35	12 25	42 875	5.9160 798	3.2710 663
36	12 96	46 656	6.	3.3019 272
37	13 69	50 653	6.0827 625	3.3322 218
38	14 44	54 872	6.1644 14	3.3619 754
39	15 21	59 319	6.2449 98	3.3912 114
40	16 00	64 000	6.3245 553	3.4199 519
41	16 81	68 921	6.4031 242	3.4482 172

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
42	17 64	74 088	6.4807 407	3.4760 266
43	18 49	79 507	6.5574 385	3.5033 981
44	19 36	85 184	6.6332 496	3.5303 483
45	20 25	91 125	6.7082 039	3.5568 933
46	21 16	97 336	6.7823 3	3.5830 479
47	22 09	103 823	6.8556 546	3.6088 261
48	23 04	110 592	6.9282 032	3.6342 411
49	24 01	117 649	7.	3.6593 057
50	25 00	125 000	7.0710 678	3.6840 314
51	26 01	132 651	7.1414 284	3.7084 298
52	27 04	140 608	7.2111 026	3.7325 111
53	28 09	148 877	7.2801 099	3.7562 858
54	29 16	157 464	7.3484 692	3.7797 631
55	30 25	166 375	7.4161 985	3.8029 525
56	31 36	175 616	7.4833 148	3.8258 624
57	32 49	185 193	7.5498 344	3.8485 011
58	33 64	195 112	7.6157 731	3.8708 766
59	34 81	205 379	7.6811 457	3.8929 965
60	36 00	216 000	7.7459 667	3.9148 676
61	37 21	226 981	7.8102 497	3.9364 972
62	38 44	238 328	7.8740 079	3.9578 915
63	39 69	250 047	7.9372 539	3.9790 571
64	40 96	262 144	8.	4.
65	42 25	274 625	8.0622 577	4.0207 256
66	43 56	287 496	8.1240 384	4.0412 401
67	44 89	300 763	8.1853 528	4.0615 48
68	46 24	314 432	8.2462 113	4.0816 551
69	47 61	328 509	8.3066 239	4.1015 661
70	49 00	343 000	8.3666 003	4.1212 853
71	50 41	357 911	8.4261 498	4.1408 178
72	51 84	373 248	8.4852 814	4.1601 676
73	53 29	389 017	8.5440 037	4.1793 39
74	54 76	405 224	8.6023 253	4 1983 364
75	56 25	421 875	8.6602 54	4.2171 633
76	57 76	438 976	8.7177 979	4.2358 236
77	59 29	456 533	8.7749 644	4.2543 21
78	60 84	474 552	8.8317 609	4.2726 586
79	62 41	493 039	8.8881 944	4.2908 404
80	64 00	512 000	8.9442 719	4.3088 695
81	65 61	531 441	9.	4.3267 487
82	67 24	551 368	9.0553 851	4.3444 815
83	68 89	571 787	9.1104 336	4.3620 707

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
84	70 56	592 704	9.1651 514	4.3795 191
85	72 25	614 125	9.2195 445	4.3968 296
86	73 96	636 056	9.2736 185	4.4140 049
87	75 69	658 503	9.3273 791	4.4310 476
88	77 44	681 472	9.3808 315	4.4479 602
89	79 21	704 969	9.4339 811	4.4647 451
90	81 00	729 000	9.4868 33	4.4814 047
91	82 81	753 571	9.5393 92	4.4979 414
92	84 64	778 688	9.5916 63	4.5143 574
93	86 49	804 357	9.6436 508	4.5306 549
94	88 36	830 584	9.6953 597	4.5468 359
95	90 25	857 375	9.7467 943	4.5629 026
96	92 16	884 736	9.7979 59	4.5788 57
97	94 09	912 673	9.8488 578	4.5947 009
98	96 04	941 192	9.8994 949	4.6104 363
99	98 01	970 299	9.9498 744	4.6260 65
100	1 00 00	1 000 000	10.	4.6415 888
101	1 02 01	1 030 301	10.0498 756	4.6570 095
102	1 04 04	1 061 208	10.0995 049	4.6723 287
103	1 06 09	1 092 727	10.1488 916	4.6875 482
104	1 08 16	1 124 864	10.1980 39	4.7026 694
105	1 10 25	1 157 625	10.2469 508	4.7176 94
106	1 12 36	1 191 016	10.2956 301	4.7326 235
107	1 14 49	1 225 043	10.3440 804	4.7474 594
108	1 16 64	1 259 712	10.3923 048	4.7622 032
109	1 18 81	1 295 029	10.4403 065	4.7768 562
110	1 21 00	1 331 000	10.4880 885	4.7914 199
111	1 23 21	1 367 631	10.5356 538	4.8058 995
112	1 25 44	1 404 928	10.5830 052	4.8202 845
113	1 27 69	1 442 897	10.6301 458	4.8345 881
114	1 29 96	1 481 544	10.6770 783	4.8488 076
115	1 32 25	1 520 875	10.7238 053	4.8629 442
116	1 34 56	1 560 896	10.7703 296	4.8769 99
117	1 36 89	1 601 613	10.8166 538	4.8909 732
118	1 39 24	1 643 032	10.8627 805	4.9048 681
119	1 41 61	1 685 159	10.9087 121	4.9186 847
120	1 44 00	1 728 000	10.9544 512	4.9324 242
121	1 46 41	1 771 561	11.	4.9460 874
122	1 48 34	1 815 848	11.0453 61	4.9596 757
123	1 51 29	1 860 867	11.0905 365	4.9731 898
124	1 53 76	1 906 624	11.1355 287	4.9866 31
125	1 56 25	1 953 125	11.1803 399	5.

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
126	1 58 76	2 000 376	11.2249 722	5.0132 979
127	1 61 29	2 048 383	11.2694 277	5.0265 257
128	1 63 84	2 097 152	11.3137 085	5.0396 842
129	1 66 41	2 146 689	11.3578 167	5.0527 743
130	1 69 00	2 197 000	11.4017 543	5.0657 97
131	1 71 61	2 248 091	11.4455 231	5.0787 531
132	1 74 24	2 299 968	11.4891 253	5.0916 434
133	1 76 89	2 352 637	11.5325 626	5.1044 687
134	1 79 56	2 406 104	11.5758 369	5.1172 299
135	1 82 25	2 460 375	11.6189 5	5.1299 278
136	1 84 96	2 515 456	11.6619 038	5.1425 632
137	1 87 69	2 571 353	11.7046 999	5.1551 367
138	1 90 44	2 628 072	11.7473 401	5.1676 493
139	1 93 21	2 685 619	11.7898 261	5.1801 015
140	1 96 00	2 744 000	11.8321 596	5.1924 941
141	1 98 81	2 803 221	11.8743 421	5.2048 279
142	2 01 64	2 863 288	11.9163 753	5.2171 034
143	2 04 49	2 924 207	11.9582 607	5.2293 215
144	2 07 36	2 985 984	12.	5.2414 828
145	2 10 25	3 048 625	12.0415 946	5.2535 879
146	2 13 16	3 112 136	12.0830 46	5.2656 374
147	2 16 09	3 176 523	12.1243 557	5.2776 321
148	2 19 04	3 241 792	12.1655 251	5.2895 725
149	2 22 01	3 307 949	12.2065 556	5.3014 592
150	2 25 00	3 375 000	12.2474 487	5.3132 928
151	2 28 01	3 442 951	12.2882 057	5.3250 74
152	2 31 04	3 511 008	12.3288 28	5.3368 033
153	2 34 09	3 581 577	12.3693 169	5.3484 812
154	2 37 16	3 652 264	12.4096 736	5.3601 084
155	2 40 25	3 723 875	12.4498 996	5.3716 854
156	2 43 36	3 796 416	12.4899 96	5.3832 126
157	2 46 49	3 869 893	12.5299 641	5.3946 907
158	2 49 64	3 944 312	12.5698 051	5.4061 202
159	2 52 81	4 019 679	12.6095 202	5.4175 015
160	2 56 00	4 096 000	12.6491 106	5.4288 352
161	2 59 21	4 173 281	12.6885 775	5.4401 218
162	2 62 44	4 251 528	12.7279 221	5.4513 618
163	2 65 69	4 330 747	12.7671 453	5.4625 556
164	2 68 96	4 410 944	12.8062 485	5.4737 037
165	2 72 25	4 492 125	12.8452 326	5.4848 066
166	2 75 56	4 574 296	12.8840 987	5.4958 647
167	2 78 89	4 657 463	12.9228 48	5.5068 784

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
168	2 82 24	4 741 632	12.9614 814	5.5178 484
169	2 85 61	4 826 809	13.	5.5287 748
170	2 89 00	4 913 000	13.0384 048	5.5396 583
171	2 92 41	5 000 211	13.0766 968	5.5504 991
172	2 95 84	5 088 448	13.1148 77	5.5612 978
173	2 99 29	5 177 717	13.1529 464	5.5720 546
174	3 02 76	5 268 024	13.1909 06	5.5827 702
175	3 06 25	5 359 375	13.2287 566	5.5934 447
176	3 09 76	5 451 776	13.2664 992	5.6040 787
177	3 13 29	5 545 233	13.3041 347	5.6146 724
178	3 16 84	5 639 752	13.3416 641	5.6252 263
179	3 20 41	5 735 339	13.3790 882	5.6357 408
180	3 24 00	5 832 000	13.4164 079	5.6462 162
181	3 27 61	5 929 741	13.4536 24	5.6566 528
182	3 31 24	6 028 568	13.4907 376	5.6670 511
183	3 34 89	6 128 487	13.5277 493	5.6774 114
184	3 38 56	6 229 504	13.5646 6	5.6877 34
185	3 42 25	6 331 625	13.6014 705	5.6980 192
186	3 45 96	6 434 856	13.6381 817	5.7082 675
187	3 49 69	6 539 203	13.6747 943	5.7184 791
188	3 53 44	6 644 672	13.7113 092	5.7286 543
189	3 57 21	6 751 269	13.7477 271	5.7387 936
190	3 61 00	6 859 000	13.7840 488	5.7488 971
191	3 64 81	6 967 871	13.8202 75	5.7589 652
192	3 68 64	7 077 888	13.8564 065	5.7689 982
193	3 72 49	7 189 057	13.8924 4	5.7789 966
194	3 76 36	7 301 384	13.9283 883	5.7889 604
195	3 80 25	7 414 875	13.9642 4	5.7988 9
196	3 84 16	7 529 536	14.	5.8087 857
197	3 88 09	7 645 373	14.0356 688	5.8186 479
198	3 92 04	7 762 392	14.0712 473	5.8284 867
199	3 96 01	7 880 599	14.1067 36	5.8382 725
200	4 00 00	8 000 000	14.1421 356	5.8480 355
201	4 04 01	8 120 601	14.1774 469	5.8577 66
202	4 08 04	8 242 408	14.2126 704	5.8674 673
203	4 12 09	8 365 427	14.2478 068	5.8771 307
204	4 16 16	8 489 664	14.2828 569	5.8867 653
205	4 20 25	8 615 125	14.3178 211	5.8963 685
206	4 24 36	8 741 816	14.3527 001	5.9059 406
207	4 28 49	8 869 743	14.3874 946	5.9154 817
208	4 32 64	8 998 912	14.4222 051	5.9249 921
209	4 36 81	9 129 329	14.4568 323	5.9344 721

TABLE—(Continued)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
210	4 41 00	9 261 000	14.4913 767	5.9439 22
211	4 45 21	9 393 931	14.5258 39	5.9533 418
212	4 49 44	9 528 128	14.5602 198	5.9627 32
213	4 53 69	9 663 597	14.5945 195	5.9720 926
214	4 57 96	9 800 344	14.6287 388	5.9814 24
215	4 62 25	9 938 375	14.6628 783	5.9907 264
216	4 66 56	10 077 696	14.6969 385	6.
217	4 70 89	10 218 313	14.7309 199	6.0092 45
218	4 75 24	10 360 232	14.7648 231	6.0184 617
219	4 79 61	10 503 459	14.7986 486	6.0276 502
220	4 84 00	10 648 000	14.8323 97	6.0368 107
221	4 88 41	10 793 861	14.8660 687	6.0459 435
222	4 92 84	10 941 048	14.8996 644	6.0550 489
223	4 97 29	11 089 567	14.9331 845	6.0641 27
224	5 01 76	11 239 424	14.9666 295	6.0731 779
225	5 06 25	11 390 625	15.	6.0822 02
226	5 10 76	11 543 176	15.0332 964	6.0911 994
227	5 15 29	11 697 083	15.0665 192	6.1001 702
228	5 19 84	11 852 352	15.0996 689	6.1091 147
229	5 24 41	12 008 989	15.1327 46	6.1180 332
230	5 29 00	12 167 000	15.1657 509	6.1269 257
231	5 33 61	12 326 391	15.1986 842	6.1357 924
232	5 38 24	12 487 168	15.2315 462	6.1446 337
233	5 42 89	12 649 337	15.2643 375	6.1534 495
234	5 47 56	12 812 904	15.2970 585	6.1622 401
235	5 52 25	12 977 875	15.3297 097	6.1710 058
236	5 56 96	13 144 256	15.3622 915	6.1797 466
237	5 61 69	13 312 053	15.3948 043	6.1884 628
238	5 66 44	13 481 272	15.4272 486	6.1971 544
239	5 71 21	13 651 919	15.4596 248	6.2058 218
240	5 76 00	13 824 000	15.4919 334	6.2144 65
241	5 80 81	13 997 521	15.5241 747	6.2230 843
242	5 85 64	14 172 488	15.5563 492	6.2316 797
243	5 90 49	14 348 907	15.5884 573	6.2402 515
244	5 95 36	14 526 784	15.6204 994	6.2487 998
245	6 00 25	14 706 125	15.6524 758	6.2573 248
246	6 05 16	14 886 936	15.6843 871	6.2658 266
247	6 10 09	15 069 223	15.7162 336	6.2743 054
248	6 15 04	15 252 992	15.7480 157	6.2827 613
249	6 20 01	15 438 249	15.7797 338	6.2911 946
250	6 25 00	15 625 000	15.8113 883	6.2996 053
251	6 30 01	15 813 251	15.8429 795	6.3079 935

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
252	6 35 04	16 003 008	15.8745 079	6.3163 596
253	6 40 09	16 194 277	15.9059 737	6.3247 035
254	6 45 16	16 387 064	15.9373 775	6.3330 256
255	6 50 25	16 581 375	15.9687 194	6.3413 257
256	6 55 36	16 777 216	16.	6.3496 042
257	6 60 49	16 974 593	16.0312 195	6.3578 611
258	6 65 64	17 173 512	16.0623 784	6.3660 968
259	6 70 81	17 373 979	16.0934 769	6.3743 111
260	6 76 00	17 576 000	16.1245 155	6.3825 043
261	6 81 21	17 779 581	16.1554 944	6.3906 765
262	6 86 44	17 984 728	16.1864 141	6.3988 279
263	6 91 69	18 191 447	16.2172 747	6.4069 585
264	6 96 96	18 399 744	16.2480 768	6.4150 687
265	7 02 25	18 609 625	16.2788 206	6.4231 583
266	7 07 56	18 821 096	16.3095 064	6.4312 276
267	7 12 89	19 034 163	16.3401 346	6.4392 767
268	7 18 24	19 248 832	16.3707 055	6.4473 057
269	7 23 61	19 465 109	16.4012 195	6.4553 148
270	7 29 00	19 683 000	16.4316 767	6.4633 041
271	7 34 41	19 902 511	16.4620 776	6.4712 736
272	7 39 84	20 123 648	16.4924 225	6.4792 236
273	7 45 29	20 346 417	16.5227 116	6.4871 541
274	7 50 76	20 570 824	16.5529 454	6.4950 653
275	7 56 25	20 796 875	16.5831 24	6.5029 572
276	7 61 76	21 024 576	16.6132 477	6.5108 3
277	7 67 29	21 253 933	16.6433 17	6.5186 839
278	7 72 84	21 484 952	16.6783 32	6.5265 189
279	7 78 41	21 717 639	16.7032 931	6.5343 351
280	7 84 00	21 952 000	16.7332 005	6.5421 326
281	7 89 61	22 188 041	16.7630 546	6.5499 116
282	7 95 24	22 425 768	16.7928 556	6.5576 722
283	8 00 89	22 665 187	16.8226 038	6.5654 144
284	8 06 56	22 906 304	16.8522 995	6.5731 385
285	8 12 25	23 149 125	16.8819 43	6.5808 443
286	8 17 96	23 393 656	16.9115 345	6.5885 323
287	8 23 69	23 639 903	16.9410 743	6.5962 023
288	8 29 44	23 887 872	16.9705 627	6.6038 545
289	8 35 21	24 137 569	17.	6.6114 89
290	8 41 00	24 389 000	17.0293 864	6.6191 06
291	8 46 81	24 642 171	17.0587 221	6.6267 054
292	8 52 64	24 897 088	17.0880 075	6.6342 874
293	8 58 49	25 153 757	17.1172 428	6.6418 522

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
294	8 64 36	25 412 184	17.1464 282	6.6493 998
295	8 70 25	25 672 375	17.1755 64	6.6569 302
296	8 76 16	25 934 336	17.2046 505	6.6644 437
297	8 82 09	26 198 073	17.2336 879	6.6719 403
298	8 88 04	26 463 592	17.2626 765	6.6794 2
299	8 94 01	26 730 899	17.2916 165	6.6868 831
300	9 00 00	27 000 000	17.3205 081	6.6943 295
301	9 06 01	27 270 901	17.3493 516	6.7017 593
302	9 12 04	27 543 608	17.3781 472	6.7091 729
303	9 18 09	27 818 127	17.4068 952	6.7165 7
304	9 24 16	28 094 464	17.4355 958	6.7239 508
305	9 30 25	28 372 625	17.4642 492	6.7313 155
306	9 36 36	28 652 616	17.4928 557	6.7386 641
307	9 42 49	28 934 443	17.5214 155	6.7459 967
308	9 48 64	29 218 112	17.5499 288	6.7533 134
309	9 54 81	29 503 609	17.5783 958	6.7606 143
310	9 61 00	29 791 000	17.6068 169	6.7678 995
311	9 67 21	30 080 231	17.6151 921	6.7751 69
312	9 73 44	30 371 328	17.6635 217	6.7824 229
313	9 79 69	30 664 297	17.6918 06	6.7896 613
314	9 85 96	30 959 144	17.7200 451	6.7968 844
315	9 92 25	31 255 875	17.7482 393	6.8040 921
316	9 98 56	31 554 496	17.7763 888	6.8112 847
317	10 04 89	31 855 013	17.8044 938	6.8184 62
318	10 11 24	32 157 432	17.8325 545	6.8256 242
319	10 17 61	32 461 759	17.8605 711	6.8327 714
320	10 24 00	32 768 000	17.8885 438	6.8399 037
321	10 30 41	33 076 161	17.9164 729	6.8470 213
322	10 36 84	33 386 248	17.9443 584	6.8541 24
323	10 43 29	33 698 267	17.9722 008	6.8612 12
324	10 49 76	34 012 224	18.	6.8682 855
325	10 56 25	34 328 125	18.0277 564	6.8753 433
326	10 62 76	34 645 976	18.0554 701	6.8823 888
327	10 69 29	34 965 783	18.0831 413	6.8894 188
328	10 75 84	35 287 552	18.1107 703	6.8964 345
329	10 82 41	35 611 289	18.1383 571	6.9034 359
330	10 89 00	35 937 000	18.1659 021	6.9104 232
331	10 95 61	36 264 691	18.1934 054	6.9173 964
332	11 02 24	36 594 368	18.2208 672	6.9243 556
333	11 08 89	36 926 037	18.2482 876	6.9313 088
334	11 15 56	37 259 704	18.2756 669	6.9382 321
335	11 22 25	37 595 375	18.3030 052	6.9451 496

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
336	11 28 96	37 933 056	18.3303 028	6.9520 533
337	11 35 69	38 272 753	18.3575 598	6.9589 434
338	11 42 44	38 614 472	18.3847 763	6.9658 198
339	11 49 21	38 958 219	18.4119 526	6.9726 826
340	11 56 00	39 304 000	18.4390 889	6.9795 321
341	11 62 81	39 651 821	18.4661 853	6.9863 681
342	11 69 64	40 001 688	18.4932 42	6.9931 906
343	11 76 49	40 353 607	18.5202 592	7.
344	11 83 36	40 707 584	18.5472 37	7.0067 962
345	11 90 25	41 063 625	18.5741 756	7.0135 791
346	11 97 16	41 421 736	18.6010 752	7.0203 49
347	12 04 09	41 781 923	18.6279 36	7.0271 058
348	12 11 04	42 144 192	18.6547 581	7.0338 497
349	12 18 01	42 508 549	18.6815 417	7.0405 806
350	12 25 00	42 875 000	18.7082 869	7.0472 987
351	12 32 01	43 243 551	18.7349 94	7.0540 041
352	12 39 04	43 614 208	18.7616 63	7.0606 967
353	12 46 09	43 986 977	18.7882 942	7.0673 767
354	12 53 16	44 361 864	18.8148 877	7.0740 44
355	12 60 25	44 738 875	18.8414 437	7.0806 988
356	12 67 36	45 118 016	18.8679 623	7.0873 411
357	12 74 49	45 499 293	18.8944 436	7.0939 709
358	12 81 64	45 882 712	18.9208 879	7.1005 885
359	12 88 81	46 268 279	18.9472 953	7.1071 937
360	12 96 00	46 656 000	18.9736 66	7.1137 866
361	13 03 21	47 045 831	19.	7.1203 674
362	13 10 44	47 437 928	19.0262 976	7.1269 36
363	13 17 69	47 832 147	19.0525 589	7.1334 925
364	13 24 96	48 228 544	19.0787 84	7.1400 37
365	13 32 25	48 627 125	19.1049 732	7.1465 695
366	13 39 56	49 027 896	19.1311 265	7.1530 901
367	13 46 89	49 430 863	19.1572 441	7.1595 988
368	13 54 24	49 836 032	19.1833 261	7.1660 957
369	13 61 61	50 243 409	19.2093 727	7.1725 809
370	13 69 00	50 653 000	19.2353 841	7.1790 544
371	13 76 41	51 064 811	19.2613 603	7.1855 162
372	13 83 84	51 478 848	19.2873 015	7.1919 663
373	13 91 29	51 895 117	19.3132 079	7.1984 05
374	13 98 76	52 313 624	19.3390 796	7.2048 322
375	14 06 25	52 734 375	19.3649 167	7.2112 479
376	14 13 76	53 157 376	19.3907 194	7.2176 522
377	14 21 29	53 582 633	19.4164 878	7.2240 45

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
378	14 28 84	54 010 152	19.4422 221	7.2304 268
379	14 36 41	54 439 939	19.4679 223	7.2367 972
380	14 44 00	54 872 000	19.4935 887	7.2431 565
381	14 51 61	55 306 341	19.5192 213	7.2495 045
382	14 59 24	55 742 968	19.5448 203	7.2558 415
383	14 66 89	56 181 887	19.5703 858	7.2621 675
384	14 74 56	56 623 104	19.5959 179	7.2684 824
385	14 82 25	57 066 625	19.6214 169	7.2747 864
386	14 89 96	57 512 456	19.6468 827	7.2810 794
387	14 97 69	57 960 603	19.6723 156	7.2873 617
388	15 05 44	58 411 072	19.6977 156	7.2936 33
389	15 13 21	58 863 869	19.7230 829	7.2998 936
390	15 21 00	59 319 000	19.7484 177	7.3061 436
391	15 28 81	59 776 471	19.7737 199	7.3123 828
392	15 36 64	60 236 288	19.7989 899	7.3186 114
393	15 44 49	60 698 457	19.8242 276	7.3248 295
394	15 52 36	61 162 984	19.8494 332	7.3310 369
395	15 60 25	61 629 875	19.8746 069	7.3372 339
396	15 68 16	62 099 136	19.8997 487	7.3434 205
397	15 76 09	62 570 773	19.9248 588	7.3495 966
398	15 84 04	63 044 792	19.9499 373	7.3557 624
399	15 92 01	63 521 199	19.9749 844	7.3619 178
400	16 00 00	64 000 000	20.	7.3680 63
401	16 08 01	64 481 201	20.0249 844	7.3741 979
402	16 16 04	64 964 808	20.0499 377	7.3803 227
403	16 24 09	65 450 827	20.0748 599	7.3864 373
404	16 32 16	65 939 264	20.0997 512	7.3925 418
405	16 40 25	66 430 125	20.1246 118	7.3986 363
406	16 48 36	66 923 416	20.1494 417	7.4047 206
407	16 56 49	67 419 143	20.1742 41	7.4107 95
408	16 64 64	67 917 312	20.1990 099	7.4168 595
409	16 72 81	68 417 929	20.2237 484	7.4229 142
410	16 81 00	68 921 000	20.2484 567	7.4289 589
411	16 89 21	69 426 531	20.2731 349	7.4349 938
412	16 97 44	69 934 528	20.2977 831	7.4410 189
413	17 05 69	70 444 997	20.3224 014	7.4470 342
414	17 13 96	70 957 944	20.3469 899	7.4530 399
415	17 22 25	71 473 375	20.3715 488	7.4590 359
416	17 30 56	71 991 296	20.3960 781	7.4650 223
417	17 38 89	72 511 713	20.4205 779	7.4709 991
418	17 47 24	73 034 632	20.4450 483	7.4769 664
419	17 55 61	73 560 059	20.4694 895	7.4829 242

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
420	17 64 00	74 088 000	20.4939 015	7.4888 724
421	17 72 41	74 618 461	20.5182 845	7.4948 113
422	17 80 84	75 151 448	20.5426 386	7.5007 406
423	17 89 29	75 686 967	20.5669 638	7.5066 607
424	17 97 76	76 225 024	20.5912 603	7.5125 715
425	18 06 25	76 765 625	20.6155 281	7.5184 73
426	18 14 76	77 308 776	20.6397 674	7.5243 652
427	18 23 29	77 854 483	20.6639 783	7.5302 482
428	18 31 84	78 402 752	20.6881 609	7.5361 221
429	18 40 41	78 953 589	20.7123 152	7.5419 867
430	18 49 00	79 507 000	20.7364 414	7.5478 423
431	18 57 61	80 062 991	20.7605 395	7.5536 888
432	18 66 24	80 621 568	20.7846 097	7.5595 263
433	18 74 89	81 182 737	20.8086 52	7.5653 548
434	18 83 56	81 746 504	20.8326 667	7.5711 743
435	18 92 25	82 312 875	20.8566 536	7.5769 849
436	19 00 96	82 881 856	20.8806 13	7.5827 865
437	19 09 69	83 453 453	20.9045 45	7.5885 793
438	19 18 44	84 027 672	20.9284 495	7.5943 633
439	19 27 21	84 604 519	20.9523 268	7.6001 385
440	19 36 00	85 184 000	20.9761 77	7.6059 049
441	19 44 81	85 766 121	21.	7.6116 626
442	19 53 64	86 350 888	21.0237 96	7.6174 116
443	19 62 49	86 938 307	21.0475 652	7.6231 519
444	19 71 36	87 528 384	21.0713 075	7.6288 837
445	19 80 25	88 121 125	21.0950 231	7.6346 067
446	19 89 16	88 716 536	21.1187 121	7.6403 213
447	19 98 09	89 314 623	21.1423 745	7.6460 272
448	20 07 04	89 915 392	21.1660 105	7.6517 247
449	20 16 01	90 518 849	21.1896 201	7.6574 138
450	20 25 00	91 125 000	21.2132 034	7.6630 943
451	20 34 01	91 733 851	21.2367 606	7.6687 665
452	20 43 04	92 345 408	21.2602 916	7.6744 303
453	20 52 09	92 959 677	21.2837 967	7.6800 857
454	20 61 16	93 576 664	21.3072 758	7.6857 328
455	20 70 25	94 196 375	21.3307 29	7.6913 717
456	20 79 36	94 818 816	21.3541 565	7.6970 023
457	20 88 49	95 443 993	21.3775 583	7.7026 246
458	20 97 64	96 071 912	21.4009 346	7.7082 388
459	21 06 81	96 702 579	21.4242 853	7.7138 448
460	21 16 00	97 336 000	21.4476 106	7.7194 426
461	21 25 21	97 972 181	21.4709 106	7.7250 325

TABLE—(*Continued*)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
462	21 34 44	98 611 128	21.4941 853	7.7306 141
463	21 43 69	99 252 847	21.5174 348	7.7361 877
464	21 52 96	99 897 344	21.5406 592	7.7417 532
465	21 62 25	100 544 625	21.5638 587	7.7473 109
466	21 71 56	101 194 696	21.5870 331	7.7528 606
467	21 80 89	101 847 563	21.6101 828	7.7584 023
468	21 90 24	102 503 232	21.6333 077	7.7639 361
469	21 99 61	103 161 709	21.6564 078	7.7694 62
470	22 09 00	103 823 000	21.6794 834	7.7749 801
471	22 18 41	104 487 111	21.7025 344	7.7804 904
472	22 27 84	105 154 048	21.7255 61	7.7859 928
473	22 37 29	105 823 817	21.7485 632	7.7914 875
474	22 46 76	106 496 424	21.7715 411	7.7969 745
475	22 56 25	107 171 875	21.7944 947	7.8024 538
476	22 65 76	107 850 176	21.8174 242	7.8079 254
477	22 75 29	108 531 333	21.8403 297	7.8133 892
478	22 84 84	109 215 352	21.8632 111	7.8188 456
479	22 94 41	109 902 239	21.8860 686	7.8242 942
480	23 04 00	110 592 000	21.9089 023	7.8297 353
481	23 13 61	111 284 641	21.9317 122	7.8351 688
482	23 23 24	111 980 168	21.9544 984	7.8405 949
483	23 32 89	112 678 587	21.9772 61	7.8460 134
484	23 42 56	113 379 904	22.	7.8514 244
485	23 52 25	114 084 125	22.0227 155	7.8568 281
486	23 61 96	114 791 256	22.0454 077	7.8622 242
487	23 71 69	115 501 303	22.0680 765	7.8676 13
488	23 81 44	116 214 272	22.0907 22	7.8729 944
489	23 91 21	116 930 169	22.1133 444	7.8783 684
490	24 01 00	117 649 000	22.1359 436	7.8837 352
491	24 10 81	118 370 771	22.1585 198	7.8890 946
492	24 20 64	119 095 488	22.1810 73	7·8944 468
493	24 30 49	119 823 157	22.2036 033	7.8997 917
494	24 40 36	120 553 784	22.2261 108	7.9051 294
495	24 50 25	121 287 375	22.2485 955	7.9104 599
496	24 60 16	122 023 936	22.2710 575	7.9157 832
497	24 70 09	122 763 473	22.2934 968	7.9210 994
498	24 80 04	123 505 992	22.3159 136	7.9264 085
499	24 90 01	124 251 499	22.3383 079	7.9317 104
500	25 00 00	125 000 000	22.3606 798	7.9370 053
501	25 10 01	125 751 501	22.3830 293	7.9422 931
502	25 20 04	126 506 008	22.4053 565	7.9475 739
503	25 30 09	127 263 527	22.4276 615	7.9528 477

TABLE—(Continued)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
504	25 40 16	128 024 064	22.4499 443	7.9581 144
505	25 50 25	128 787 625	22.4722 051	7.9633 743
506	25 60 36	129 554 246	22.4944 438	7.9686 271
507	25 70 49	130 323 843	22.5166 605	7.9738 731
508	25 80 64	131 096 512	22.5388 553	7.9791 122
509	25 90 81	131 872 229	22.5610 283	7.9843 444
510	26 01 00	132 651 000	22.5831 796	7.9895 697
511	26 11 21	133 432 831	22.6053 091	7.9947 883
512	26 21 44	134 217 728	22.6274 17	8.
513	26 31 69	135 005 697	22.6495 033	8.0052 049
514	26 41 96	135 796 744	22.6715 681	8.0104 032
515	26 52 25	136 590 875	22.6936 114	8.0155 946
516	26 62 56	137 388 096	22.7156 334	8.0207 794
517	26 72 89	138 188 413	22.7376 340	8.0259 574
518	26 83 24	138 991 832	22.7596 134	8.0311 287
519	26 93 61	139 798 359	22.7815 715	8.0362 935
520	27 04 00	140 608 000	22.8035 085	8.0414 515
521	27 14 41	141 420 761	22.8254 244	8.0466 03
522	27 24 84	142 236 648	22.8473 193	8.0517 479
523	27 35 29	143 055 667	22.8691 933	8.0568 862
524	27 45 76	143 877 824	22.8910 463	8.0620 18
525	27 56 25	144 703 125	22.9128 785	8.0671 432
526	27 66 76	145 531 576	22.9346 899	8.0722 62
527	27 77 29	146 363 183	22.9564 806	8.0773 743
528	27 87 84	147 197 952	22.9782 506	8.0824 8
529	27 98 41	148 035 889	23.	8.0875 794
530	28 09 00	148 877 000	23.0217 289	8.0926 723
531	28 19 61	149 721 291	23.0434 372	8.0977 589
532	28 30 24	150 568 768	23.0651 252	8.1028 39
533	28 40 89	151 419 437	23.0867 928	8.1079 128
534	28 51 56	152 273 304	23.1084 4	8.1129 803
535	28 62 25	153 130 375	23.1300 67	8.1180 414
536	28 72 96	153 990 656	23.1516 738	8.1230 962
537	28 83 69	154 854 153	23.1732 605	8.1281 447
538	28 94 44	155 720 872	23.1948 37	8.1331 87
539	29 05 21	156 590 819	23.2163 735	8.1382 23
540	29 16 00	157 464 000	23.2379 001	8.1432 529
541	29 26 81	158 340 421	23.2594 067	8.1482 765
542	29 37 64	159 220 088	23.2808 935	8.1532 939
543	29 48 49	160 103 007	23.3023 604	8.1583 051
544	29 59 36	160 989 184	23.3238 076	8.1633 102
545	29 70 25	161 878 625	23.3452 351	8.1683 092

TABLE—(Continued)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
546	29 81 16	162 771 336	23.3666 429	8.1733 02
547	29 92 09	163 667 323	23.3880 311	8.1782 888
548	30 03 04	164 566 592	23.4093 998	8.1832 695
549	30 14 01	165 469 149	23.4307 49	8.1882 441
550	30 25 00	166 375 000	23.4520 788	8.1932 127
551	30 36 01	167 284 151	23.4733 892	8.1981 753
552	30 47 04	168 196 608	23.4946 802	8.2031 319
553	30 58 09	169 112 377	23.5159 52	8.2080 825
554	30 69 16	170 031 464	23.5372 046	8.2130 271
555	30 80 25	170 953 875	23.5584 38	8.2179 657
556	30 91 36	171 879 616	23.5796 522	8.2228 985
557	31 02 49	172 808 693	23.6008 474	8.2278 254
558	31 13 64	173 741 112	23.6220 236	8.2327 463
559	31 24 81	174 676 879	23.6431 808	8.2376 614
560	31 36 00	175 616 000	23.6643 191	8.2425 706
561	31 47 21	176 558 481	23.6854 386	8.2474 74
562	31 58 44	177 504 328	23.7065 392	8.2523 715
563	31 69 69	178 453 547	23.7276 21	8.2572 635
564	31 80 96	179 406 144	23.7486 842	8.2621 492
565	31 92 25	180 362 125	23.7697 286	8.2670 294
566	32 03 56	181 321 496	23.7907 545	8.2719 039
567	32 14 89	182 284 263	23.8117 618	8.2767 726
568	32 26 24	183 250 432	23.8327 506	8.2816 255
569	32 37 61	184 220 009	23.8537 209	8.2864 928
570	32 49 00	185 193 000	23.8746 728	8.2913 444
571	32 60 41	186 169 411	23.8956 063	8.2961 903
572	32 71 84	187 149 248	23.9165 215	8.3010 304
573	32 83 29	188 132 517	23.9374 184	8.3058 651
574	32 94 76	189 119 224	23.9582 971	8.3106 941
575	33 06 25	190 109 375	23.9791 576	8.3155 175
576	33 17 76	191 102 976	24.	8.3203 353
577	33 29 29	192 100 033	24.0208 243	8.3251 475
578	33 40 84	193 100 552	24.0416 306	8.3299 542
579	33 52 41	194 104 539	24.0624 188	8.3347 553
580	33 64 00	195 112 000	24.0831 891	8.3395 509
581	33 75 61	196 122 941	24.1039 416	8.3443 41
582	33 87 24	197 137 368	24.1246 762	8.3491 256
583	33 98 89	198 155 287	24.1453 929	8.3539 047
584	34 10 56	199 176 704	24.1660 919	8.3586 784
585	34 22 25	200 201 625	24.1867 732	8.3634 466
586	34 33 96	201 230 056	24.2074 369	8.3682 095
587	34 45 69	202 262 003	24.2280 829	8.3729 668

TABLE—(Concluded)

OF SQUARES, CUBES, AND SQUARE AND CUBE ROOTS, ETC.

Number.	Square.	Cube.	Square Root.	Cube Root.
588	34 57 44	203 297 472	24.2487 113	8.3777 188
589	34 69 21	204 336 469	24.2693 222	8.3824 653
590	34 81 00	205 379 000	24.2899 156	8.3872 065
591	34 92 81	206 425 071	24.3104 916	8.3919 423
592	35 04 64	207 474 688	24.3310 501	8.3966 729
593	35 16 49	208 527 857	24.3515 913	8.4013 981
594	35 28 36	209 584 584	24.3721 152	8.4061 180
595	35 40 25	210 644 875	24.3926 218	8.4108 326
596	35 52 16	211 708 736	24.4131 112	8.4155 419
597	35 64 09	212 776 173	24.4335 834	8.4202 46
598	35 76 04	213 847 192	24.4540 385	8.4249 448
599	35 88 01	214 921 799	24.4744 765	8.4296 383
600	36 00 00	216 000 000	24.4948 974	8.4343 267
601	36 12 01	217 081 801	24.5153 013	8.4390 098
602	36 24 04	218 167 208	24.5356 883	8.4436 877
603	36 36 09	219 256 227	24.5560 583	8.4483 605
604	36 48 16	220 348 864	24.5764 115	8.4530 281
605	36 60 25	221 445 125	24.5967 478	8.4576 906
606	36 72 36	222 545 016	24.6170 673	8.4623 479
607	36 84 49	223 648 543	24.6373 7	8.467
608	36 96 64	224 755 712	24.6576 56	8.4716 471
609	37 08 81	225 866 529	24.6779 254	8.4762 892
610	37 21 00	226 981 000	24.6981 781	8.4809 261
611	37 33 21	228 099 131	24.7184 142	8.4855 579
612	37 45 44	229 220 928	24.7386 338	8.4901 848
613	37 57 69	230 346 397	24.7588 368	8.4948 065
614	37 69 96	231 475 544	24.7790 234	8.4994 233
615	37 82 25	232 608 375	24.7991 935	8.5040 35
616	37 94 56	233 744 896	24.8193 473	8.5086 417
617	38 06 89	234 885 113	24.8394 847	8.5132 435
618	38 19 24	236 029 032	24.8596 058	8.5178 403
619	38 31 61	237 176 659	24.8797 106	8.5224 321
620	38 44 00	238 328 000	24.8997 992	8.5270 189
621	38 56 41	239 483 061	24.9199	8.5316
622	38 68 84	240 641 848	24.9399	8.5362
623	38 81 29	241 804 367	24.9600	8.5408
624	38 93 76	242 970 624	24.9800	8.5453
625	39 06 25	244 140 625	25.	8.5499
626	39 18 76	245 314 376	25.0200	8.5544
627	39 31 29	246 491 883	25.0400	8.5590
628	39 43 84	247 673 152	25.0599	8.5635
629	39 56 41	248 858 189	25.0799	8.5681

TABLE

SHOWING THE TENSILE STRENGTH OF VARIOUS QUALITIES OF WROUGHT-IRON.

American Wrought-Iron.

	Breaking weight of a square inch bar.
From Salisbury, Conn	58,000
" " "	66,000
" Pittsfield, Mass	57,000
" Bellefonte, Pa	58,000
" Maramec, Mo	43,000
" " "	53,000
" Centre County, Pa	58,400
" Lancaster County, Pa	58,061
" Carp River, Lake Superior	89,582
" Mountain, Mo., charcoal bloom	90,000
American, hammered	53,900
Chain-iron	43,000
Rivets	53,300
Bolts	52,250
Boiler-plates	50,000
" "	60,000
Average boiler-plates	55,000
" joints, double-riveted	35,000
" " single "	28,600

English and other Wrought-Irons.

Iron, English bar	56,000
" mean of English	53,900
" rivets	65,000
Lowmoor iron	56,100
" " plates	57,881
Bowling plates	53,488
Glasgow best boiler	56,317
" ship-plates	53,870
Yorkshire plates	57,724
Staffordshire plates	43,821
Derbyshire plates	48,563
Bessemer wrought-iron	65,253
" " "	76,195

		Breaking weight of a square inch bar.
Bessemer wrought-iron		82,110
Russian " "		59,500
" " "		76,084
Swedish " "		58,084

TABLE

SHOWING THE ACTUAL EXTENSION OF WROUGHT-IRON AT VARIOUS TEMPERATURES.

Deg. of Fah.	Length.	
32°	1·	
212	1·0011356	
392	1·0025757	Surface becomes straw-colored, deep yellow, crimson, violet, purple, deep blue, bright blue.
672	1·0043253	
752	1·0063894	
932	1·0087730	Surface becomes dull, and then bright red.
1112	1·0114811	
1652	1·0216024	Bright red, yellow, welding heat, white heat.
2192	1·0348242	
2732	1·0512815	
2912	Cohesion destroyed. Fusion perfect.	

Linear Expansion of Wrought-Iron.—The linear expansion a bar of wrought-iron undergoes, according to Daniell's pyrometer, when heated from the freezing- to the boiling-point, or from 32° to 212° Fah., is about $\frac{1}{880}$ of its length; at higher temperatures, the elongation becomes more rapid.

Thus, it will be seen how sensible a change takes place when iron undergoes a variation of temperature. A bar of iron, 10 feet long, subject to an ordinary change of temperature of from 32° to 180° Fah., will elongate more than $\frac{1}{8}$ of an inch, or sufficient to cause fracture in stone work, strip the thread of a screw, or endanger a bridge, floor, roof, or trusses.

The expansion of volume and surface of wrought-iron is calculated by taking the linear expansion as unity; then, following the geometrical law, the superficial expan-

sion is twice the linear, and the cubical expansion is three times the linear.

Wrought-Iron will bear on a square inch, without permanent alteration, 17,800 pounds, and an extension in length of $\frac{1}{1400}$. Cohesive force is diminished $\frac{1}{3000}$ by an increase of 1 degree of heat.

Compared with cast-iron, its strength is 1·12 times, its extensibility 0·86 times, and its stiffness 1·3 times.

TABLE

SHOWING THE TENSILE STRENGTH OF VARIOUS QUALITIES OF STEEL PLATES.

	Breaking weight of a square inch bar.
Mersey Co., puddled steel	108,906
" " ship-plates	99,468
Blochairn puddled steel	106,394
" boiler-plates	89,447
Naylor, Vickers & Co., cast	87,972
" " " "	95,196
T. Turton & Son	95,360
Moss & Gamble's	81,588
Shortridge, Howell & Co.	108,900
Homogeneous metal	105,732
" " 2d quality	81,662
Bessemer steel	148,324
" "	154,825
" "	157,881
American chrome steel, highest strength	198,910
" " " lowest "	163,760
" " " average "	180,000

TABLE

SHOWING THE TENSILE STRENGTH OF VARIOUS QUALITIES OF CAST-IRON.

American Cast-Iron.

	Breaking weight of a square inch bar.
Common pig-iron	15,000
Good common castings	20,000

	Breaking weight of a square inch bar.
Cast-iron castings	20,834
" "	19,200
" "	27,700
Gun-heads, specimen from	24,000
" " "	39,500
Greenwood cast-iron	21,300
" " (after third melting)	45,970
Mean of American cast-iron	31,829
Gun-metal, mean	37,232

English Cast-Iron.

Lowmoor	14,076
Clyde, No. 1	16,125
Clyde, No. 3	23,468
Calder, No. 1	13,735
Stirling, mean	25,764
Mean of English	19,484
Stirling, toughened iron	28,000
Carron, No. 2, cold-blast	16,683
" " 2, hot-blast	13,505
" " 3, cold-blast	13,200
" " 3, hot-blast	17,755
Davon, No. 3, hot-blast	21,907
Buffery, No. 1, cold-blast	17,466
" " 1, hot-blast	13,437
Cold-Talon (North Wales), No. 2, cold-blast	18,855
" " " " " 2, hot-blast	16,676

Cast-Iron expands $\frac{1}{162000}$ of its length for 1 degree of heat; the greatest change in the shade, in this climate, is $\frac{1}{1170}$ of its length; exposed to the sun's rays, $\frac{1}{1000}$.

Cast-iron shrinks, in cooling, from $\frac{1}{85}$ to $\frac{1}{98}$ of its length.

Cast-iron is crushed by a force of 93,000 pounds upon a square inch, and will bear, without permanent alteration, 15,300 pounds upon a square inch.

TABLE

SHOWING THE WEIGHT OF BOILER-PLATES 1 FOOT SQUARE, AND FROM $\frac{1}{16}$TH TO AN INCH THICK.

Thickness in inches..............	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{15}{16}$	1
Weight in lbs. per foot sq......	$2\frac{1}{2}$	5	$7\frac{1}{2}$	10	$12\frac{1}{2}$	15	$17\frac{1}{2}$	20	$22\frac{1}{2}$	25	$27\frac{1}{2}$	30	$32\frac{1}{2}$	35	$37\frac{1}{2}$	40

TABLE

SHOWING THE WEIGHT OF SQUARE BAF-IRON, FROM $\frac{1}{2}$ AN INCH TO 6 INCHES SQUARE, 1 FOOT LONG.

Square...........	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2	$2\frac{1}{8}$	$2\frac{1}{4}$	$2\frac{3}{8}$	$2\frac{1}{2}$	$2\frac{5}{8}$	$2\frac{3}{4}$	$2\frac{7}{8}$	3	$3\frac{1}{8}$
Weight in lbs...	$\frac{8}{10}$	$1\frac{1}{4}$	2	$2\frac{1}{2}$	$3\frac{1}{2}$	$4\frac{1}{4}$	$5\frac{1}{4}$	$6\frac{1}{2}$	$7\frac{1}{2}$	9	$10\frac{1}{2}$	12	$13\frac{1}{2}$	$15\frac{1}{4}$	17	19	22	$23\frac{1}{4}$	$25\frac{1}{2}$	28	$30\frac{1}{2}$	33

Square.........	$3\frac{1}{4}$	$3\frac{3}{8}$	$3\frac{1}{2}$	$3\frac{5}{8}$	$3\frac{3}{4}$	$3\frac{7}{8}$	4	$4\frac{1}{8}$	$4\frac{1}{4}$	$4\frac{3}{8}$	$4\frac{1}{2}$	$4\frac{5}{8}$	$4\frac{3}{4}$	$4\frac{7}{8}$	5	$5\frac{1}{4}$	$5\frac{1}{2}$	$5\frac{3}{4}$	6
Weight in lbs.	$35\frac{3}{4}$	$38\frac{1}{2}$	$41\frac{1}{2}$	$44\frac{1}{2}$	$47\frac{1}{2}$	$50\frac{3}{4}$	54	$57\frac{1}{2}$	61	$64\frac{3}{4}$	$68\frac{1}{2}$	$72\frac{1}{4}$	$76\frac{1}{4}$	$80\frac{1}{4}$	$84\frac{1}{2}$	$93\frac{1}{4}$	$102\frac{1}{4}$	$111\frac{3}{4}$	$121\frac{3}{4}$

TABLE

SHOWING THE WEIGHT OF ROUND-IRON FROM ½ AN INCH TO 6 INCHES DIAMETER, 1 FOOT LONG.

For Calculating the Weight of Shafting, etc.

Diameter in inches..............	½	⅝	¾	⅞	1	1⅛	1¼	1⅜	1½	1⅝	1¾	1⅞	2	2⅛
Weight in pounds................	¾	1	1½	2	2¾	3½	4¼	5	6	7	8	9½	10½	12

Diameter in inches..............	2¼	2⅜	2½	2⅝	2¾	2⅞	3	3⅛	3¼	3⅜	3½	3⅝	3¾	3⅞
Weight in pounds................	13½	15	16¾	18¼	20	22	24	26	28	30¼	32½	35	37¼	40

Diameter in inches.	4	4⅛	4¼	4⅜	4½	4⅝	4¾	4⅞	5	5¼	5½	5¾	6
Weight in pounds................	42½	45¼	48	50¾	53¾	56¾	60	63	66¾	73¼	80¼	87¾	95½

TABLES

SHOWING THE WEIGHT OF CAST-IRON BALLS FROM 3 TO 13 INCHES IN DIAMETER.

Diameter in inches..	3	3¼	3½	3¾	4	4¼	4½	4¾	5	5¼	5½	5¾	6	6¼	6½	6¾	7	7¼	7½	7¾
Weight in pounds....	3¾	4¾	5¾	7¼	8¾	10½	12½	14¾	17	20	23	26	29¾	33½	37¾	42¼	47¼	52½	58	64

Diameter in inches..	8	8¼	8½	8¾	9	9¼	9½	9¾	10	10¼	10½	10¾	11	11½	12	13
Weight in pounds....	70½	77¼	84¼	92½	100¼	109	118	127½	137¾	148¼	159½	171	183¼	209½	238	302

WEIGHT OF CAST-IRON PLATES PER SUPERFICIAL FOOT AS PER THICKNESS.

Thickness in inches.........	⅛	¼	⅜	½	⅝	¾	⅞	1
Weight........................	lbs. oz. 4 13	lbs. oz. 9 10	lbs. oz. 14 8	lbs. oz. 19 5	lbs. oz. 24 2	lbs. oz. 29 0	lbs. oz. 33 13	lbs. oz. 38 10

TABLE

SHOWING THE WEIGHT OF CAST-IRON PIPES, 1 FOOT IN LENGTH, FROM ¼ INCH TO 1¼ INCHES THICK, AND FROM 3 TO 24 INCHES DIAMETER.

Diameter of Bore in Inches.	Thickness in Inches.								
	¼ Lbs.	⅜ Lbs.	½ Lbs.	⅝ Lbs.	¾ Lbs.	⅞ Lbs.	1 Lbs.	1⅛ Lbs.	1¼ Lbs.
3	8½	12½	17¼	22¼	27½	....	....	....	....
3½	9¼	14¼	19½	25¼	31¼	....	....	....	....
4	10	16¾	22	28½	35	....	....	....	....
4½	11¾	18	2[illegible]½	3[illegible]½	38¾	....	....	....	....
5	13	19¾	27	34½	42¼	50½	59	....	....
5½	15	21½	29½	37½	46	54¾	63¾	....	....
6	....	23½	32	40¾	49¾	59	68¾	78¾	88¾
6½	....	25¼	34½	43¾	53½	63½	73½	84¼	95
7	....	27¼	36¾	46¾	56¾	67¾	78½	89¾	101¼
7½	....	29	39	50	60¾	72	83½	95¼	107½
8	....	30¾	41¾	53	64½	76¼	88½	100¾	113½
8½	....	33	44½	56¼	68¼	80¾	93½	106½	120
9	....	34½	46½	59	71¾	84¾	98½	111¾	125¾
9½	....	36¼	49	62	75½	89	103	117½	132
10	....	38¼	51½	65¼	79¼	93½	108	122¾	138
10½	....	....	54	68¼	82¾	97¾	112¾	128½	144¼
11	....	....	56½	71¼	86½	102	117¾	134	150¼
11½	....	....	59	76¼	90	106¼	122¾	139½	156½
12	....	....	61¼	77½	93½	110½	127½	145	162½
13	....	....	....	82¾	101¼	118¼	137½	154	173½
14	....	....	....	89¼	108¼	126½	146¼	165¼	185¼
15	....	....	....	95¼	115¾	135¼	156¼	176¼	198
16	....	....	....	....	123¼	143	166	187½	211¼
17	....	....	....	....	130¼	152½	178½	198¼	223½
18	....	....	....	....	137	161¼	185¼	209	235¼
19	....	....	....	....	....	169¼	195¾	222¼	247
20	....	....	....	....	....	178	205¼	233¼	259
21	....	....	....	....	....	....	214	243½	273¼
22	....	....	....	....	....	....	223½	244¾	285¼
23	....	....	....	....	....	....	233½	265½	298¼
24	....	....	....	....	....	....	245¼	277½	310½

TABLE

SHOWING THE WEIGHT PER SQUARE FOOT OF WROUGHT-IRON, STEEL, COPPER, AND BRASS.

Thickness advancing by 100th of an inch.	Weight Wrought-iron per square foot.	Weight Steel per square foot.	Weight Copper per square foot.	Weight Brass per square foot.
.01	.4049	.4087	.4625	.4367
.02	.8098	.8174	.9250	.8736
.03	1.2147	1.2261	1.3875	1.3104
.04	1.6196	1.6348	1.8500	1.7472
.05	2.0245	2.0435	2.3125	2.1840
.06	2.4294	2.4522	2.7750	2.6208
.07	2.8343	2.8609	3.3375	3.0576
.08	3.2392	3.2696	3.7000	3.4944
.09	3.6441	3.6783	4.1624	3.9312
.10	4.0490	4.0870	4.6250	4.3680
.11	4.4539	4.4957	5.0875	4.8048
.12	4.8588	4.9044	5.5500	5.2416
.13	5.2637	5.3131	6.0125	5.6784
.14	5.6686	5.7218	6.4750	6.1152
.15	6.0735	6.1305	6.9375	6.5520
.16	6.4784	6.5392	7.4000	6.9888
.17	6.8833	6.9479	7.8625	7.4256
.18	7.2882	7.3566	8.3250	7.8624
.19	7.6931	7.7653	8.7875	8.2992
.20	8.0980	8.1740	9.2500	8.7368
.21	8.5029	8.5827	9.7125	9.1728
.22	8.9078	8.9914	10.1750	9.6096
.23	9.3127	9.4001	10.6375	10.0464
.24	9.7176	9.8088	11.1000	10.4832
.25	10.1225	10.2175	11.5625	10.9200
.26	10.5274	10.6262	12.0250	11.3568
.27	10.9323	11.0349	12.4875	11.7936
.28	11.3372	11.4436	12.9500	12.2304
.29	11.7421	11.8523	13.4125	12.6673
.30	12.1470	12.2610	13.8750	13.1040
.31	12.5519	12.6697	14.3375	13.5408
.32	12.9568	13.0784	14.8000	13.9776
.33	13.3617	13.4871	15.2625	14.4144
.34	13.7666	13.8958	15.7250	14.8512
.35	14.1715	14.3045	16.1775	15.2880

TABLE—(*Continued*)

SHOWING THE WEIGHT PER SQUARE FOOT OF WROUGHT-IRON, STEEL, COPPER, AND BRASS.

Thickness advancing by 100th of an inch.	Weight Wrought-iron per square foot.	Weight Steel per square foot.	Weight Copper per square foot.	Weight Brass per square foot.
.36	14.5764	14.7132	16.6500	15.7248
.37	14.9813	15.1219	17.1125	16.1616
.38	15.3862	15.5306	17.5750	16.5984
.39	15.6911	15.9393	18.0375	17.0352
.40	16.1960	16.3480	18.5000	17.4720
.41	16.6009	16.7567	18.9625	17.9088
.42	17.0058	17.1654	19.4250	18.3446
.43	17.4107	17.5741	19.8875	18.7824
.44	17.8156	17.9828	20.3590	19.2192
.45	18.2205	18.3915	20.8125	19.6560
.46	18.6254	18.8002	21.2750	20.0928
.47	19.0303	19.2089	21.7375	20.5296
.48	19.4352	19.6176	22.2000	20.9664
.49	19.8401	20.0263	22.6625	21.4032
.50	20.2450	20.4350	23.1250	21.8400
.51	20.6499	20.8437	23.5875	22.2768
.52	21.0548	21.2524	24.0500	22.7136
.53	21.4597	21.6611	24.5125	23.1504
.54	21.8646	22.0698	24.9750	23.5872
.55	22.2695	22.4785	25.4375	24.0240
.56	22.6744	22.8872	25.9900	24.4608
.57	23.0793	23.2959	26.3625	24.8976
.58	23.4842	23.7046	26.8250	25.3344
.59	23.8891	24.1133	27.2875	25.7712
.60	24.2940	24.5220	27.7500	26.2080
.61	24.6989	24.9307	28.2125	26.6448
.62	25.1038	25.3394	28.6750	27.0816
.63	25.5087	25.7481	29.1375	27.5184
.64	25.9136	26.1568	29.6000	27.9552
.65	26.3185	26.5655	30.0625	28.3920
.66	26.7234	26.9742	30.5250	28.8288
.67	27.1283	27.2829	30.9875	29.2656
.68	27.5332	27.7916	31.4500	29.7024
.69	27.9381	28.3003	31.9125	30.1392
.70	28.3430	28.6090	32.3750	30.5760

TABLE—(*Concluded*)

SHOWING THE WEIGHT PER SQUARE FOOT OF WROUGHT-IRON, STEEL, COPPER, AND BRASS.

Thickness advancing by 100th of an inch.	Weight Wrought-iron per square foot.	Weight Steel per square foot.	Weight Copper per square foot.	Weight Brass per square foot.
.71	28.7479	29.0177	32.8375	31.0128
.72	29.1528	29.4264	33.3000	31.4496
.73	29.5577	29.8351	33.7625	31.8864
.74	29.9626	30.2438	34.2250	32.3232
.75	30.3675	30.6525	34.6875	32.7600
.76	30.7724	31.0612	35.1500	33.1968
.77	31.1773	31.4699	35.6125	33.6336
.78	31.5822	31.8786	36.0750	34.0704
.79	31.9871	32.2873	36.5375	34.5072
.80	32.3920	32.6960	37.0000	34.9440
.81	32.7969	33.1047	37.4625	35.3808
.82	33.2018	33.5134	37.9250	35.8176
.83	33.6067	33.9221	38.3875	36.2544
.84	34.0116	34 3308	38.8500	36.6912
.85	34.4165	34.7395	39.3125	37.1280
.86	34.8214	35.1482	39.7750	37.5648
.87	35.2263	35.5569	40.2375	38.0016
.88	35.6312	35.9656	40.7000	38.4384
.89	36.0361	36.3743	41.1625	38.8752
.90	36.4410	36.7830	41.6250	39.3120
.91	36.8459	37.1917	42.0875	39.7488
.92	37.2508	37.6004	42.5500	40.1856
.93	37.6557	38.0091	43.0125	40.6224
.94	38.0606	38.4178	43.4750	41.0592
.95	38.4655	38.8265	43.9375	41.4960
.96	38.8704	39.2352	44.4000	41.9328
.97	39.2753	39.6439	44.8625	42.3696
.98	39.6802	40.0526	45.3250	42.8064
.99	40.0851	40.4613	45.7875	43.2432
100	40.4900	40.8700	46.2500	43.6800

RULES FOR FINDING THE DIAMETER AND SPEED OF PULLEYS.

To find the Size of Driving-Pulley.—Multiply the diameter of the driven by the number of revolutions it should make, and divide the product by the revolutions of the driver. The quotient will be the size of the driver.

EXAMPLE.

Diameter of driven, 20 inches,	198
Number of revolutions driven, 198,	20
" " " of driver, 220,	220)3960
	18 inch.

The Diameter and Revolutions of Driver being given, to find the Diameter of the Driven that shall make a given Number of Revolutions.—Multiply the diameter of the driver by its number of revolutions, and divide the product by the number of revolutions of the driven. The quotient will be the size of the driven.

EXAMPLE.

Diameter of driver, 18 inches,	220
Number of revolutions of driver, 220,	18
" " " " driven, 198,	198)3960
	20 inch.

To find the Number of Revolutions of the Driven Pulley.—Multiply the diameter of the driver by its number of revolutions, and divide by diameter of driven. The quotient will be the number of revolutions of the driven.

EXAMPLE.

Diameter of driver, 18 inches,	220
Number of revolutions of driver, 220,	18
Diameter of driven, 20,	20)3960
	198 revolutions.

GEARING.

Gearing is the general term employed to denote a combination of mechanical organs, interposed between the prime mover and the working parts of machinery. Frequently, however, the signification is restricted to the series of toothed wheels by which the motion is conducted from one revolving axis to another, independently of the shafts and bearings by which they are supported. Two toothed wheels are also said to gear when they have their teeth engaged together; and to be out of gear, when separate, and consequently out of action.

Rule *for finding the Diameter of Toothed Wheels.*—Multiply the number of teeth by the number of thirty-seconds of an inch contained in the pitch, the product will be the diameter in inches and hundredths of an inch; or, multiply the number of teeth by the true pitch, and the product by ·3184. These results give only the diameter between the pitch-line on one side, and the same line on the other side, and not the entire diameter from point to point of teeth on opposite sides. It must also be borne in mind that these results are only approximate diameters, since the wheel often varies from the computed diameter in consequence of shrinkage and other causes.

Rule *for finding the Required Number of Teeth in a Pinion to have any given Velocity.*—Multiply the velocity or number of revolutions of the driver by its number of teeth or its diameter, and divide the product by the desired number of revolutions of the pinion or driven.

Rule *for finding the Diameter of a Pinion, when the Diameter of the Driver and the Number of Teeth in Driver and Pinion are given.*—Multiply the diameter of driver by the number of teeth in the pinion, and divide the pro-

duct by the number of teeth in the driver, and the quotient will be the diameter of pinion.

Rule *for finding the Number of Revolutions of a Pinion or Driven, when the Number of Revolutions of Driver and the Diameter or the Number of Teeth of Driver and Driven are given.*—Multiply the number of revolutions of driver by its number of teeth or its diameter, and divide the product by the number of teeth or the diameter of the driven.

Rule *for finding the Number of Revolutions of a Driver, when the Revolutions of Driven and Teeth, or Diameter of Driver and Driven, are given.*—Multiply the number of teeth or the diameter of driven by its revolutions, and divide the product by the number of teeth or the diameter of driver.

Rule *for finding the Number of Revolutions of the last Wheel at the end of a train of Spur-Wheels, all of which are in a line, and mesh into one another, when the Revolutions of the First Wheel, and the Number of Teeth, or the Diameter of the First and Last are given.*—Multiply the revolutions of first wheel by its number of teeth or its diameter, and divide the product by the number of teeth or the diameter of the last wheel; the result is its number of revolutions.

Rule *for finding the Number of Revolutions in each Wheel for a train of Spur-Wheels, each to have a given Velocity.*—Multiply the number of revolutions of the driving-wheel by its number of teeth, and divide the product by the number of revolutions each wheel is to make. The results will be the number of teeth required for each.

Rule *for finding the Number of Revolutions of the Last Wheel in a train of Wheels and Pinions, Spurs or Bevels, when the Revolutions of the First or Driver, and the Diam-*

eter, the Teeth or the Circumference of all the Drivers and Pinions are given.—Multiply the diameter, the circumference, or the number of teeth of all the driving-wheels together, and this continued product by the number of revolutions of the first wheel; and divide this product by the continued product of the diameter, the circumference, or the number of teeth of all the pinions, and the quotient will be the number of revolutions of the last wheel.

BELTING.

While the use of belts for the transmission of power is not an American invention, the numerous improvements made in this country have caused it to be known in Europe as the American system. In Europe, the greater part of the power is transmitted by cog-wheels, but in this country, 99 per cent. is transmitted by belting. The latter is used everywhere, from the sewing-machine to the 500-horse-power engine of the largest factory.

Belts can be run in any way, at any angle, of any length, and at any speed, and can be put up by any one of ordinary skill. They can be made of any flexible material—leather, rubber, gutta-percha, or cloth; yet, while so handy and so popular, they have one fault—they are not positive. If the motor makes a certain number of revolutions, a portion of them are lost with every belt used. This is the only fault of the system. It is noiseless, yielding, and regular; but, unlike cog-wheels, it is not positive. The number of revolutions that are lost may, and do, vary continually by changes of the load, or of the atmosphere.

Belts derive their power to transmit motion from the friction between the surface of the belt and the pulley, and from nothing else, and are governed by the same laws as in friction between flat surfaces. The friction increases regularly with the pressure. The great difference often

observed in the friction of belts is due simply to their elasticity of surface; that is, the more elastic the surface, the greater the friction.

Very careful experiments have been made, both in this country and in France, in regard to the transmitting power of flat leather belts, and the agreement of the results by independent investigators entitle them to great confidence. These experiments show how much power a belt can be expected to transmit with safety when various data are given; but they do not show how much power can be transmitted by a belt.

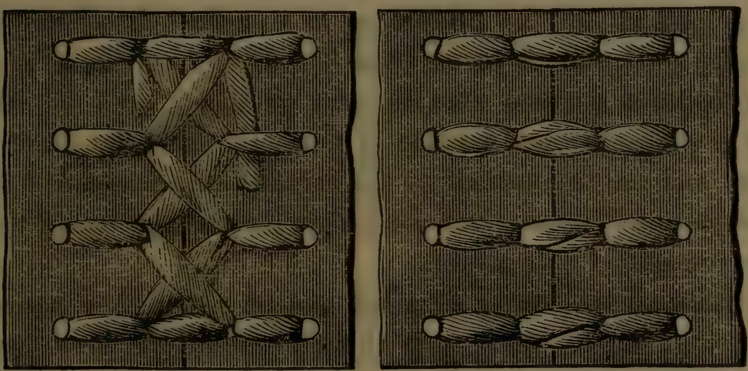
Take, for instance, two belts of the same width, the same condition, and the same quality of material, one of them is run so slack that it continually sags or flaps, while the other is strained so tight that it frequently breaks; now, if both belts be run at the same speed, the tight belt will transmit nearly twice as much power as the slack one. Now, if the actual strain on both belts were known, and also the diameter and condition of the pulleys, the amount of power transmitted could be calculated with considerable accuracy.

Search may be made in vain for any such data, as the general rule for calculating the power a belt will transmit seems to be that an inch belt, with a speed of 1000 feet per minute, will transmit a horse-power, without taking into account the tension of the belt, diameter of pulleys, etc., points which have been determined by experiment to affect the question materially.

This seems to confirm the statement that power transmitted by belts is ordinarily estimated by guesswork, and also that the guesses are quite as likely to be wrong as right; therefore, it is not to be wondered at that there should be numerous quarrels and lawsuits between landlords and their tenants in regard to the power used by the latter.

On the scientific principle that the adhesion, and consequently the capability of leather belts to transmit power from motors to machines, is in proportion to the pressure of the actual weight of the leather on the surface of the pulley, it is manifest that, as longer belts have more weight than shorter ones, and that broader belts of the same length have more weight than narrower ones, it may be adopted as a rule that the adhesion and capability of belts to transmit power, is in the ratio of their relative lengths and breadths.

A belt of double the length or breadth of another, under the same circumstances, will be found capable of transmitting double the power. For this reason it is desirable to use long belts. By doubling the velocity of the same belt, its effectual capability for transmitting power is also doubled.



Improved methods of Lacing Belts.

It is a common error, among mechanics and owners of factories, to make the face of their pulleys narrow, in order to economize on the first cost of pulleys and belting; but this false economy seldom decreases the cost of pulleys, and only saves a trifle in the first cost of belting. The small amount saved is soon lost by the stopping of

machinery caused by the slipping of belts, strain on the shafting, increased friction, requiring additional driving-power, and rapid destruction of the belts themselves.

Leather belts used with grain side to the pulley, will not only do more work, but last longer than if used with flesh side to the pulley. This is owing to the fact that the grain side is more compact and fixed than the flesh side, and more of its surface is brought in contact with the pulley.

The smoother the two surfaces, the less air will pass between the belt and the pulley. The more uneven the surface of the belt and pulley, the more strain necessary to prevent the belt slipping; for what is lost by want of contact, must be made up by extra strain on the belt.

Leather belts, with grain side to pulley, can drive 34 per cent. more than flesh side; for this reason, in all cases where the face of pulleys are not turned and polished, they should be covered with leather.

A belt should be pliable, so as to adjust itself readily to the pulley as it passes over it. If the belt be pliable, its contact surface smooth and polished, and it runs on a pulley whose surface is perfectly smooth, there is obtained the full benefit of the atmospheric pressure upon the belt as it passes over the pulley, minus the centrifugal force caused by its motion.

Belts should never be oiled except when they become dry and hard; and then the oil should be used very sparingly, as it not only rots the leather, but causes the belt to stretch.

In oiling or greasing a belt, avoid everything of a pasty nature. The belt should be made pliable, not covered with a sticky substance.

Horizontal Belts.—The driving half of horizontal belts should be the lower half, when practicable, as, when the belt stretches, the upper half will cover more of the

pulley's surface. Long horizontal belts are better than short ones, as their weight increases their contact with the pulley.

Perpendicular Belts.—Belts running on pulleys perpendicular to each other, should be kept tightly strained, as their weight tends to decrease their contact with the lower pulleys.

In putting on a new belt or taking up an old one, great care should be taken to have the ends perfectly square, and the lace- or hook-holes punched exactly opposite to each other. Many fail in these respects, and in consequence have crooked belts.

A good leather belt, one inch wide, has sufficient strength to lift 1000 pounds.

The speed of a mile per minute, for main driving leather belts, has been found both safe and advantageous for practical use.

The capability of belts to transmit power is determined by the extent of their adhesion to the surface of pulleys.

The extent of the adhesion of belts varies greatly under the varying circumstances in the use of them, and is limited in comparison with the absolute strength of the leather.

The adhesion and friction causing the belt to cling to the surface of a pulley without slipping, is mainly governed by the weight of the leather—if used horizontally.

If belts are strained tightly on the pulleys, then the adhesion is increased in proportion to the increased tension produced.

Double Leather Belts.—Double leather belts are frequently used; but it is clearly a mistake, as a single leather one will transmit more of the power than a double one. Double leather belts run straighter than single ones, as the flank side of one part can be put against the back of

the others. A double belt will stand a greater tension than a single one, but a single one will stand all that should be put upon any belt.

Rubber Belts.—A rubber belt will transmit as much power as a leather belt with the same tension, and will last as long, and run perfectly straight. It can be made of any length or width, of exactly the same thickness in every part, perfectly smooth on its surface, and, when in use, every part will come in contact with the face of the pulley. The greater tractile power of a rubber belt is due to its surface elasticity.

Rule *for finding Length of Belt wanted.*—Add the diameter of the two pulleys together; divide the sum by 2 and multiply the quotient by $3\frac{1}{4}$. Add the product to twice the distance between the centres of the shafts, and the sum will be the length required.

Rule *for finding the Width of Belt to Transmit a given Horse-power.*—Multiply 36,000 by the number of horse-power. Multiply the speed of the belt in feet per minute by one-half the length in inches of belt in contact with smaller pulley. Divide the first product by the second. The quotient will be the required width in inches.

Another Rule *for finding the Width of Belt, etc.*—Multiply 36,000 by the number of horse-powers; divide the product by the number of feet the belt is to travel per minute; now, divide the quotient by the number of feet, or parts of feet, of belt in contact with the smaller pulley; divide this last quotient by 6, and the result is the required width of belt in inches.

Rule *for finding the Change required in the Length of a Belt when one of the Pulleys on which it runs, is changed for one of a Different Size.*—Take three times the difference between the diameters of the pulleys, and divide by 2. The result will be the length of belt to cut out or put in.

Rule *for calculating the Number of Horse-powers a Belt will Transmit; its Velocity, and the Number of Square Inches in contact with the Smaller Pulley being given.*—Divide the number of square inches in contact with the pulley by 2; multiply this quotient by the velocity of the belt in feet per minute, and divide by 36,000. The quotient is the number of horse-powers the belt will transmit.

How to Test the Quality of Leather for Belting.—Cut a small strip of the leather about $\frac{1}{16}$ of an inch in thickness, and place it in strong vinegar. If the leather has been thoroughly tanned, and is of good quality, it will remain for months even, immersed, without alteration, simply becoming a little darker in color. But, on the contrary, if not thoroughly tanned, the fibres will quickly swell, and after a short period become transformed into a gelatinous mass.

How to Make Belts Run on the Centre of Pulleys.—It is a common occurrence for belts to run on one side of the pulleys. This arises from one or two causes: 1st, one or both of the pulleys may be conical, and of course the belt will run on the higher side. The most effectual remedy for this would be to straighten the face of the pulleys. 2d, the shafts may not be parallel, or exactly in line. In this case, the belt would incline off to the side where the ends of the shafts came nearest together. The remedy in this case, would be to slack up on the hanger-bolts, and drive the hangers out or in, as the case may be, until both ends of the shaft become exactly parallel. This can be determined by getting the centres of the shafts at both ends, by means of a long lathe or light strip of board.

Tighteners.—The tightener should be placed as close to the large or driving pulley as circumstances will permit, as the loss of power incurred by the use of the tightener is equal to that required to bend the belt and carry the

tightening pulley. Consequently, there is a greater loss of power by placing it near the small pulley, as the belt is required to be bent more than when it is placed near the large one.

CEMENT FOR MAKING STEAM-JOINTS AND PATCHING STEAM-BOILERS.

Take a quantity of pure red lead, put it in an iron mortar, or on a block, or thick plate of iron. Put in a quantity of white lead ground in oil; knead them together until you make a thick putty, then pound it; the more it is pounded, the softer it will become. Roll in red lead and pound again; repeat the operation, adding red lead and pounding until the mass becomes a good stiff putty. In applying it to the flange or joint, it is well to put a thin grummet around the orifice of the pipe to prevent the cement being forced inward to the pipe, when the bolts are screwed up.

When the flanges are not faced, make the above mass rather soft, and add cast-iron borings run through a fine sieve, when it will be found to resist either fire or water.

Another Cement.—Take 10 pounds of ground litharge, 4 pounds of ground Paris white, ½ pound of yellow ochre, and ½ ounce of hemp, cut into lengths of ½ inch; mix all together with boiled linseed-oil, to the consistency of a stiff putty. This cement resists fire, and will set in water.

Cement for Rust-Joints.—Cast-iron borings or turnings, 19 pounds; pulverized sal-ammoniac, 1 pound; flour of sulphur, ½ pound. Should be thoroughly mixed and passed through a tolerably fine sieve. Sufficient water should be added to wet the mixture through. It should be prepared some hours before being used. A small

quantity of sludge from the trough of a grinding-stone will improve its quality.

NON-CONDUCTORS FOR STEAM-PIPES AND STEAM-CYLINDERS.

A very cheap and efficient non-conductor for covering steam-pipes and cylinders may be prepared in the following proportions:—100 pounds of fire or potter's clay, dissolved in water and thoroughly mixed with 100 pounds of finely-sieved coal ashes, and $1\frac{1}{2}$ pounds of hair. After being well kneaded together, it should be allowed to stand a few hours in a damp place. Just before being used, 100 pounds of plaster of Paris should be added, which, being thoroughly mixed, may be laid on the surface to be covered, in a thin coat, with a trowel, and when dry another may be added, and so on until the desired thickness is attained, after which the surface should be made as smooth as possible. It can then be whitewashed or painted any desired color, and made to look quite ornamental. The pipe or cylinder should be warm when the non-conductor is applied.

HOW TO MARK ENGINEERS' OR MACHINISTS' TOOLS.

Any engineer or machinist can mark his name on his tools by first warming the tool to be marked, and rubbing on a thin layer of beeswax or tallow until it flows, and then letting it cool; after which the name may be marked with a dull scriber or a piece of hard wood sharpened to a point; then, by applying some nitric acid for a few minutes, the name will be found etched on the steel; after which the acid must be thoroughly washed off with water, and the tool again heated in order to remove the wax or tallow, and rubbed over with a soft rag.

TO POLISH BRASS.

Engineers will find the following recipe a good one for polishing the brass work of their engines: Oxalic acid, dissolved in rain- or cistern-water, in the proportion of about five cents' worth to a pint of water, if applied with a rag or piece of waste, will remove the tarnish from brass, and render it bright; the surface should then be rubbed with an oily rag and dried, and afterwards burnished with chalk, whiting, or rotten-stone. This is probably one of the quickest known methods for cleaning brass. A mixture of muriatic acid and alum, dissolved in water, imparts a golden color to brass articles that are steeped in it for a few seconds.

Owing to irregularities of surface, it often happens that considerable difficulty is encountered in putting a good polish on articles of brass or copper. If, however, they be immersed in a bath composed of aquafortis, 1 part; spirits of salt, 6 parts; and water, 2 parts, for a few minutes, if small, or 20 or 30 if large, they will become covered with a kind of black mud, which, on removal by rinsing, displays a beautiful lustrous under-surface. Should the lustre be deemed insufficient, the immersion may be repeated, care always being taken to rinse thoroughly. All articles cleaned in this manner should be dried in hot, dry sawdust.

SOLDER.

The following solder will braze steel, and may be found very useful in case of a valve-stem, or other light portion, breaking, when it is important that the engine should continue work for some time longer: Silver, 19 parts; copper, 1 part; brass, 2 parts. If practicable, charcoal dust should be strewed over the melted metal of the crucible.

TABLE

SHOWING WEIGHT OF DIFFERENT MATERIALS.

	Weight of cubic feet in lbs.	Weight of a cubic inch in ozs.	Number of cubic inches in a lb.	Weight of a cubic inch in lbs.
Mercury	848	7.851	2.037	.4908
Lead	709	6.456	2.437	.4103
Wrought-iron	477	4.140	3.623	.276
Cast-iron	454½	4.203	3.802	.263
Sheet Copper	557¼	5.159	3.103	.3225
Cast Copper	549¼	5.086	3.146	.3178
Cast Brass	524¾	4.852	3.223	.3037
Brick	125	1.456	13.824	.0723
Stone	151	1.396	11.443	.0873
Water	62½	0.579	27.50	.0362

JOINTS.

Rust-joints, composed of sal-ammoniac, iron borings, flour of sulphur, and water, were formerly employed for all the permanent joints around engines; but they are fast going out of use and being replaced by faced joints.

Red lead joints were also very generally used, but they are now obsolete, and justly so, not only for their dirty appearance, but also for the difficulty experienced in starting them, as it required, in most cases, the use of sledges and chisels, which incurred the danger of breaking the flanges.

All movable joints of the best description of land and marine engines are now faced on a lathe or planer, and then rendered perfectly steam-, air-, and water-tight, by filing and scraping, so that all that is necessary, when put together, is to oil their surfaces.

For smooth surfaces that can be conveniently calked, sheet copper, annealed by heating it to a cherry red, and then plunging it in cold water, makes a permanent joint.

Lead wire makes a very cheap, clean, and permanent joint. Copper wire also makes a very good joint; but, when convenient, it is always best to plane or turn a groove in one of the surfaces to be brought in contact.

For uniform surfaces, gauze wire-cloth, coated on either side with white or red lead paint, makes a very durable joint, particularly where it is exposed to high temperatures.

For pumps or stand-pipes in the holds of vessels, canvas well saturated on both sides with white or red lead, makes a very durable joint. Pasteboard, painted on both sides with white or red lead paint, is frequently used with good results.

Gum is now very generally used for steam- and water-joints, and is very convenient, as it requires no preparation, special tools, or much experience to use it; for rough or uneven surfaces, it is more reliable than any other material used for joints. But it has the defect of being very injurious to iron in consequence of the great quantity of sulphur it contains.

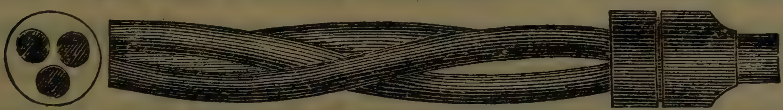
Cooling Compound for Heavy Bearings.—For cooling heavy pillow-block bearings, or the steps of upright shafts, the following will be found very valuable: 4 pounds of tallow, $\frac{1}{2}$ pound sugar of lead, and $\frac{3}{4}$ pound of plumbago. When the tallow is melted (not boiling), add the sugar of lead, and let it dissolve; then put in the plumbago. Keep stirring until the whole mass is cold.

A mixture of soft soap and black lead makes an excellent lubricant for gearing, as it lessens the abrasion and noise, and has the advantage over tallow of not becoming hard. It is also easily removed should it become necessary to clean the parts on which it has been used.

STEAM-BOILER FLUE AND TUBE CLEANERS.

Steam is more effectual and convenient than any other known method for cleaning the tubes of steam-boilers, as the operation may be easily performed while the furnace is in full blast. The steam may be taken from any convenient place by means of a gum hose, about one inch in diameter, attached to a valve by means of a nipple, the other end of the hose being securely fastened to a piece of $\frac{3}{4}$ inch gas-pipe of the desired length, which should also be furnished with a valve. This pipe should be covered with wood, which may be secured at each end by means of ferules.

To clean tubes by this method, the valve in the nozzle-pipe should be first closed; then open the supply steam and also the damper, and insert the nozzle-pipe in each tube or flue, and then turn on the steam, when all the deposit, ashes, etc., will be blown into the ash-pit. To clean the tubes of vertical boilers, the nozzle-pipe should be bent in order to make it capable of being inserted in the vertical tubes.



Pike's Boiler Tube and Flue Cleaner.

The annexed cut represents Pike's Steam-Boiler Flue and Tube Cleaner. It will be observed that the tubes of the cleaner, through which the steam escapes, are twisted into a spiral form, causing the steam to pass through the flue or tube in a spiral course, thereby effectually removing all the dust or ashes. This tube-cleaner is highly spoken of by engineers and steam users in general. It is the invention of Wm. G. Pike, an inspector of the Hartford Steam-Boiler Inspection and Insurance Company.

THE INVENTION AND IMPROVEMENT OF THE STEAM-ENGINE.

A Machine receiving at distant times and from many hands new combinations and improvements, and becoming at last of signal benefit to mankind, may be compared to a rivulet, swelled in its course by tributary streams, until it rolls along a majestic river, enriching in its progress states and provinces. In retracing the current from where it mingles with the ocean, the pretensions of even ample subsidiary streams are merged in our admiration of the master flood, glorying, as it were, in its expansion. But as we continue to ascend, those waters which nearer the sea would have been disregarded as unimportant, begin to rival in magnitude, and divide attention with the parent stream, until at length, on approaching the fountains of the river, it appears trickling from the rock, or oozing from among the flowers of the valley.

So also in developing the rise of a machine, a coarse instrument or a toy may be recognized as the germ of that production of mechanical genius whose power and usefulness have stimulated our curiosity to mark its changes and to trace its origin. The same feeling of gratitude which attached reverence to the spot from whence mighty rivers have sprung, also clothed it, as it were, with divinity, and raised altars in honor of the inventors of the saw, the plough, and the loom. To those who are familiar with modern machinery, the construction of these implements may appear to have conferred but slight claim to the respect in which their authors were held, in ancient times. Yet, artless as they seem, their use first raised man above the beasts of the field, by incalculably diminishing the sum of human labor.

From the important and increasing influence of the

steam-engine on human affairs, controversies have frequently arisen between writers of different nations respecting the claims of their countrymen to its invention. But the steam-engine cannot be said to be the invention of any one man or belong to any nationality, but to be a combination of the scattered devices of a number of ingenious men, whose fortune it was more frequently to fail than to succeed, but who did not consider such failures a good reason for abandoning their cherished objects, or allowing them to fall into oblivion, being aware that practice is progressive, and that the mechanical difficulties which so much embarrassed them would be removed as they advanced towards greater perfection, and that the schemes that had failed, as well as those that were doubtful, should be considered as seeds drifting on a common field which some random step would fix in the soil and quicken into life.

It also not unfrequently happened that some of those discoveries that conferred such benefits on mankind were the result of mere accident, and that, while in the pursuit of some peculiar objects, others of greater importance were often unfolded. Such was the case of the steam-engine, as it was only the raising of water directly by fire that exercised the ingenuity of Worcester, Moreland, Papin, Savery, and Newcomen. But their labors resulted in the production of the most important and valuable machine that the arts ever presented to man — the steam-engine.

The earliest records extant of a machine producing useful effect by the vapor of boiling water, is the Eolipile of Heron, of Alexandria, who lived about 280 years B. C., and which may be said to be the "germ" of the modern steam-engine. The Eolipile was a hollow globe resting on legs, which, being filled with water and placed over a fire, allowed the increasing steam to escape through a

small orifice at the top, which had the effect of producing a draught similar to that of the blower-pipe in the chimney of a locomotive. The Eolipile was very extensively used in Egypt for blowing fires, increasing draught in chimneys, diffusing perfumes, and for idol worship, etc.

In 1543, Blasco d'Garay, a Spaniard, is said to have constructed a steamboat of 200 tons, in the harbor of Barcelona, Spain, and used steam as a motive-power for its propulsion. From this it was claimed that the steam-engine was invented in Spain, and that Blasco d'Garay should be regarded as the inventor. But as the nature and construction of his engine are not mentioned in the claim, we are left to form our own opinion. The probabilities are that D'Garay used a machine constructed on the same principle as Savery's, Papin's, or Leopold's, to raise water upon an overshot-wheel fixed on the same axle as the paddles. D'Garay is also claimed to be the inventor of the paddle-wheel, which is evidently a mistake, as the same honor was claimed by Papin, Savery, Jouffroy, Symington, and a host of others. In fact, the principle of the paddle-wheel is equally as old as the wind-mill.

In 1630, Branca, an Italian, is claimed to have invented a machine which produced useful effect by the elastic force of steam. But from the most reliable accounts, Branca's machine was constructed on the same principle as the "breast" water-wheel, and received its motion from steam issuing from an orifice in a vessel similar to the Eolipile.

In 1663, the Marquis of Worcester is said to have constructed an engine by which motion was given to a piston by means of steam. But the account of his invention is so ambiguous as to lead to the belief that his machine was similar to those of Papin and Savery, which could not be said to belong to the same class as the modern steam-engine, nor could their inventors claim to have contributed

anything to the invention of the latter, except to make their contemporaries more familiar with the mechanical properties of steam, as their ideas seem to have been wholly confined to the raising of water in the most direct manner. There is no evidence to show that either Papin or Savery ever thought of a piston.

In 1710, Newcomen made the first steam-engine in England that could be said to be worthy of the name. To Newcomen belongs the honor of not only laying the foundation of the modern steam-engine, but also of attracting the attention of ingenious men to its improvement. Newcomen was also claimed to be the discoverer of the principle of condensation; but this is evidently a mistake, as the alchemists were familiar with the formation of a vacuum by the condensation of steam, and with raising water into it by atmospheric pressure, long before Newcomen's time.

In 1720, Leopold and Trevithick invented their high-pressure engine, which was greatly admired, though useless and impracticable, for when the steam raised the piston to the upper end of the cylinder it would remain there, as there was no counter-pressure to cause it to descend. But the engines of Newcomen and Leopold, with all their imperfections, were the connecting links between the machines of Heron, D'Garay, Papin, and Savery, and the engines of Watt, Fitch, and Oliver Evans.

In 1764, James Watt made the first engine in England that bore any resemblance to steam-engines of the modern type; and in 1786, he patented and made public his great improvements, among which was separate condensation—to realize the importance of which requires careful study and thorough mechanical knowledge even at this late day. When we consider that to him all was comparatively novel, we pause in astonishment at the stupendous results

of his invention; and yet it was eight years before he succeeded in getting any one to try it, and had not a fortunate chance at that period introduced him to a liberal, enlightened, and enterprising man in Boulton, another eight years of fruitless efforts might probably have been undergone, or even the full appreciation of the invention indefinitely delayed, in which case the whole of that vast career of progress on which the human race entered as a consequence of the steam-engine would have been postponed.

In 1787, John Fitch, of Connecticut, with the aid of a common blacksmith, built in Philadelphia the first condensing-engine ever heard of on this continent, and this without any knowledge of Watt's improvements in condensing-engines, as it was in the previous year that the latter patented and made them public — consequently, there is every reason to believe that John Fitch was entirely ignorant of them.

In 1793, Oliver Evans, a native of Philadelphia, invented the high-pressure engine, and to him must be awarded the credit of having built, and put in operation, the first practically useful high-pressure steam-engine. The high-pressure engine of Oliver Evans had immense advantages, in its cheapness and simplicity, over the more expensive and complicated condensing-engine of Watt, and ever since the days of Oliver Evans, the high-pressure engine has continued to be the standard steam-engine for land purposes wherever steam has been introduced as a motive-power. England, ever true and grateful to her own genius, has fitly honored her greatest inventor, Watt; while America has suffered the genius of Oliver Evans, John Fitch, and Robert Fulton, to die unrewarded in life, and forgotten in the grave, though she has not forgotten to profit by their inventions.

In 1807, Robert Fulton, a man whom we should never forget to honor, established the success of steam navigation. He was also the first to apply the paddle-wheel, in its present form, to the propulsion of vessels, and to introduce steam ferry-boats in this country.

It is quite interesting to follow the various improvements that have been made upon the steam-engine at different times, and to see how it has been brought to its present form. The cylinder and piston were used for raising water long before the advent of the steam-engine; and in the early forms of the latter, one end of the cylinder was open to the atmosphere, while the piston was nothing more than a flat wooden float, connected with a beam and sector by means of a rod and chain. But in 1776, Blakey made the piston steam-tight by means of a stratum of hemp saturated with grease, for which he obtained a patent.

In 1804, Oliver Evans made the cylinder a steam-tight vessel, and introduced steam alternately above and below the piston. In this arrangement lay the vital energy of the steam-engine, as all the other parts are but appendages to the cylinder and piston. They may be removed, and the energy of the machine still remains; but take away either cylinder or piston, and the whole becomes inert as the limbs of an animal whose heart has ceased to beat. The metallic piston-packing, now so universally used, was invented by Aiken in 1836, and the stuffing-box, by La Hire, in 1716.

Murdock was claimed to be the inventor of the crank, but the same device had been used in the common foot-lathe centuries before. After many years of experiment, it was finally adopted by Pickard; after which Watt patented a much more complicated method of transmitting reciprocating into rotary motion, which was called

the sun and planet motion, but it went out of use after repeated trials with the crank.

In the first steam-engines, the admission of the steam to the cylinder was regulated by means of a stop-cock, which required constant attention, as it had to be opened and closed at each stroke of the piston; but a boy, named Potter, employed in this service, stimulated by the love of play, ingeniously added cords to the levers by which the cocks were turned, and connecting the other ends of the cords to the working-beam, rendered the machine self-acting. Beighton afterwards substituted iron rods.

The parallel rods, now so universally superseded by the guides, were invented by Watt in 1790. He was also the inventor of the condenser, and the first to attach an air-pump to the steam-engine, though the latter device was used for other purposes previous to Watt's time. Watt was also the inventor of the governor; but that indispensable adjunct of the steam-engine remained very imperfect down to 1848, when George Corliss invented and constructed the first steam-engine governor that could be said to be worthy of the name.

The slide-valve was invented by Murray, in 1810. In 1832, R. L. Stevens invented the poppet-valve. In 1841, he invented the Stevens cut-off valve-gear, which is still used on a large number of marine engines. He was also the inventor of the now universally known American skeleton walking-beam, with cast-iron centre and forged straps.

In 1848, the automatic cut-off, which has almost universally superseded that relic of barbarism, the throttle-valve, was invented by George Corliss.

The combination of the foregoing devices has made up the modern steam-engine — the great prime mover of man. And, strange as it may seem, nearly all the im-

portant improvements in the steam-engine have been achieved by men of other callings than that of engineers, which goes to strengthen the often repeated assertion, that where it is possible to make any improvement in a machine, it would be more likely to be discovered by men of natural genius, untrammelled by the routine of any special trade, than by men who, from force of habit, become unreasoning creatures.

While the merit of the discovery of the expansive properties of steam is due to Hornblower, who obtained a patent for his invention in 1781, the honor of first working it expansively belongs to Robert L. Stevens, as he invented the cut-off valve in 1813, and there does not appear to be any evidence that steam was worked expansively in England previous to that time. Thus it will be seen that most of the great improvements made in the steam-engine, more particularly the high-pressure engine, were the results of American genius; and that America has produced a class of engineers who, in spite of many difficulties, have produced effects wonderful even to themselves.

Although Archimedes was the inventor, or at least the alleged inventor, of the screw, Col. John Stevens, an American, was the first to adapt it to the purposes of the propulsion of vessels, in 1804. He was also the inventor of the tubular boiler.

The spring-gauge, that invaluable attachment of the steam-boiler, is also an American invention, and with the exception of the Bourdon (French) and Schæffer (Prussian), all the spring-gauges in use in the United States, some thirty in number, are American inventions.

INDEX.

Absolute motion, 502.
zero, 316.
Accelerated motion, 503.
Acceleration, 490.
Accumulation of deposits, 459.
Actual or net horse-power, 61.
Adjustable injector, method of working the, 171.
table of capacities, 172.
Advantages of high-pressure engine, 54.
Affinity, 489.
Air, 309, 380.
and fuel, mixture of, 335.
leakage of, 261.
pressure of, 310.
pump-cover, 260.
pump, in its seat, placing the, 271.
pumps, 222.
Allen
high-pressure cut-off engine, 155, 161.
American coal and coke, table showing nature and value of varieties of, 343.
woods, table showing prominent qualities in principal, 344.
Amount of mechanical power, 288.
of resisting pressure, 201.
Analysis of anthracite, 336.
Angle, 490.
Angular advance, 106.
motion, 503.
Anthracite coal, 336.
composition of different kinds of, 336.
evaporative efficiency of a pound of, 337.
Anthracite coal, quantity of air required for combustion of, 337.
A place for everything, 6.
Application of balance-weight, 205.
Arched head, 429.
Area of a circle, rule for finding, 488.
Ashes of anthracite, 336.
Atlas Steam-pump, 178–179.
Atmosphere, 309.
column of, 232.
Atmospheric air, 45.
pressure, 32.
Attraction, capillary, 491.
Available heat of combustion, 336.
Average atmospheric pressure, 321.
Axle, 491.
Back-action engines, 208.
Balanced slide-valves, 102.
Balance-weights, application of, 205
Balancing momentum of direct-acting engines, 205.
Barometer, the, 225.
Beam-centre, relative positions of the, 268.
Beam-engine, marine, 210, 211.
Beam-engines, 209.
Beam pillow-blocks, 266.
Bedding marine boilers, 405.
Bed-plate, 74, 212, 271.
laying of the, 267.
Belting, 561.
how to test the quality of leather for, 567.
Belts, double leather, 565.
horizontal, 564.
leather, 564.
perpendicular, 565.
rubber, 565.

Belts run on the centre of pulleys, how to make, 567.
Bituminous coal, 338.
Blasco d'Garay, 576.
Bodies, density of, 321.
Boiler compound, Lord's, 450.
explosions, 452.
feed-pumps, 178–179.
flues, 417.
heads, 429.
tube- and flue-cleaner, Pike's,573.
tubular, 352.
Boilers and boiler materials, definitions applied to, 469.
cylinder, 351.
evaporative efficiency of, 372.
expansion and contraction of, 358.
flue, 352.
horse-power of, 375.
locomotive, 353.
marine, 399.
of ocean steamers, 224.
setting, 356.
steam, 350.
testing, 359.
Boiling-point for fresh water at different altitudes above sea-level, 306.
of water, 303.
Bourne's rule, 69.
Brace, 269.
Branca, 576.
Brass or composition rings, 135.
to polish, 570.
Buckeye high-pressure cut-off engine, 52, 53, 148.

Calculating weights and measures, useful numbers in, 292.
Calking, 464.
Connery's concave method of, 466.
Caloric, 321.
engine, Roper's, 159, 324, 325.
latent, 322.
radiation of, 322.
reflection of, 322.
sensible, 322.
Cam-pump, Dayton, 180–181.
Capacity, unit of, 295.
Capillary attraction, 491.
Carbon, 347.
Carbonate of magnesia, 443.
Carburetted hydrogen, 347.
Care and management of steam-boilers, 363.
of marine boilers, 406.
Cataract, the, 145.
Cement for making steam-joints and patching steam-boilers, 568.
for rust-joints, 568.
Centigrade thermometer, 315.
Central and mechanical forces and definitions, 490.
Centre, dead, 128.
Centre of gravity, 492.
of gyration, 495.
of gyration of counter-weights,205
of oscillation, 503.
Centre-points, the, 267.
Centres of the journals, 270.
Centrifugal action of screw, 277.
force, 139.
Change of zero, 316.
Chemical and mechanical remedies, 447.
equivalents, 334.
Chests, steam, 85.
Chimney, area of, rule for finding, 484.
diameter and height of, table showing proper, 484.
Chimneys, 483.
Circle, cylinder, sphere, etc., mensuration of, 487.
the, 511.
Circulation of water in boilers, effects of heat on, 331.
Cleanliness, 72.
Clearance, 99.
inside, 95.
Clock, 226.
Clothing marine boilers, 405.
Coal, anthracite, 336.
bituminous, 338.
Coal-gas, 347.
Coke, coal, and wood, table showing relative properties of, 344.
Colburn's rule, 69.

Column of atmosphere, 232.
Combustion, 332.
available heat of, 336.
spontaneous, 342.
Commercial horse-power, 62.
Common slide-valve, 94.
Communication of heat, 330.
Comparative scale of Centigrade, Fahrenheit, and Reaumer thermometers, 314.
strength of single and double riveted seams, 461.
Composition of water, 299.
Compression, 100.
and dilatation of gases, 348.
Composition of different kinds of anthracite coal, 336.
Compound engine, vertical, 199.
engines, 200.
motion, 503.
Concealed heat of steam, 26.
Concussive ebullition, 455.
Condenser, 212, 268.
jet, 216, 221.
surface, 217.
surface, Pirsson's, 218.
surface, Sewell's, 220.
Condensers, 216.
Condensing engine, working principles of, explanation of, 190.
or low-pressure steam-engine, 188.
engines, feed-pump for, 183.
engines, horse-power of, 191.
engines, power of, Watts' rule for calculating, 191.
Conditions of equilibrium, 320.
to be observed to obtain a correct diagram, 239.
Conical or poppet-valves, 95.
Connecting-rod, examination of, 110.
piston and crank connection, 80.
Connections, slide-valve, 103.
Cooling compound for heavy bearings, 572.
Corliss (Harris Corliss) high-pressure cut-off engine, 126.
high-pressure engine, 151.
valves, 125.
Corrosion of marine boilers, 451.
Counter-weights,centre of gyration of, 205.
Cover, air-pump, 260.
Crank-circle, subdivision of, 112.
Crank connection, piston, and connecting-rod, 80.
Crank, examination of principles involved in use of, 111.
of engine, 80.
Crank-pin, heating of the, 259.
table showing angular position of, 84.
Crank-pins, 83.
Cranks, 109.
Cross-head, form of, 86.
slides for the, 269.
Cross-heads, 86.
Crowding tubes, 411.
Curves, theoretical, 238.
Curvilinear seams, 470.
Cut-off engine, Allen high-pressure, 155, 161.
Buckeye high-pressure. 52, 53, 148.
Hampson and Whitehill's high-pressure, 153, 154.
Hawkins and Dodge's high-pressure, 24, 147.
Watts and Campbell's high-pressure, 37, 148.
Wheelock's high-pressure, 71.
Woodruff and Beach's high-pressure, 156.
Wright's high-pressure, 20, 146.
Cut-offs, 105, 138.
Cycloidal wheels, 283.
Cylinder boilers, 351.
bottom, the, 212, 268.
rule for finding the mean or average pressure in a, 50.
Cylinders, 74.
horizontal, 242.
oscillating, 242.
vertical, 241.

Dayton cam-pump, 180–181.
Dead centre, 128.
Decimal, 291.
arithmetic, 291.

Decimal equivalents of inches, feet, and yards, 292.
equivalents of pounds and ounces, 292.
equivalents to the fractional parts of a gallon or an inch, 294.
fractions, 291.
Defects, 457.
Definitions applied to boilers and boiler materials, 469.
Degrees of temperature, rules for comparing, 318.
Density of bodies, 321.
Deposits, accumulation of, 459.
Design of steam-engines, 73.
Deterioration of plates, 456.
D'Garay, Blasco, 576.
Diagram, conditions to be observed to obtain a correct, 239.
how to compute a, 244.
is taken, how to calculate the power the engine is exerting when the, 244.
observation on the character of, 243.
No. 1, 238.
Nos. 2 and 3, 240.
No. 4, 245.
No. 5, 247.
No. 6, 248.
Nos. 7 and 8, 249.
No. 9, 250.
No. 10, 253.
Diagrams are taken, facts to be recorded when, 242.
form of, 250.
Diameter and arrangement of tubes, 411.
and length of rock-shaft, 86.
of pump-plunger for an engine, rules for finding the, 182.
Dimensions of steam-ports, 87.
Direct-acting engines, 203.
balancing momentum of, 205.
screw-engine, 204.
Directions for setting up steam-pumps, 184.
Disconnecting paddle-wheels, 286.
Domes, steam, 353.
Double eccentrics on the shaft, formula by which to find positions of, 107.
leather belts, 565.
Draft, regulation of, 385.
Drums, mud, 355.
Dynamic equivalent of heat, 329.
Dynamics, 492.
Dynamometer, 254.
Dynamometrical horse-power, 62.
or effective horse-power of a marine engine, rules for finding the, 255.

Ebullition, concussive, 455.
Eccentric, how to find the throw of any, 106.
throw or stroke of the, 106.
Eccentric-rod, how to adjust, 109.
length of, 109.
Eccentric-rods, 108.
modes of attaching, 109.
Eccentrics, 104.
of marine engines, 106.
Economical working of steam-engine, 73.
Economy in high-pressure engines, 70.
Effective pressure against piston, 64.
Effect of heat on circulation of water in boilers, 331.
Effluent velocity of steam, 32.
Elastic fluids, 320.
Elasticity, 470; of steam, 27.
Electricity, 454.
Electro-magnetism, 72.
Energy, 492.
Engine, Harris Corliss high-pressure, 151.
crank of, 80.
Hampson and Whitehill's high-pressure, 153.
Haskin's vertical high-pressure, 163, 308.
high-pressure, 158, 319.
high-pressure, desirable, 55.
how to reverse an, 31.
in line, how to put an, 128.
Massey's rotary, 164, 333.
Naylor's vertical, 300.

Engine, Naylor's vertical high-pressure, 157.
shaft-pulley, to find diameter of 144.
steeple, 208.
vertical compound, 199.
Williams' vertical three-cylinder, 300, 319.
Engineer, the, 256.
Engineers' or machinists' tools, how to mark, 569.
Engines, back-action, 208.
beam, 209.
compound, 200.
condensing horse-power of, 191.
direct-acting, 203.
geared, 207.
in a steamboat or ship, how to put the, 265.
knocking in, 167.
land and marine, management of, 258.
oscillating, 205.
portable, 166.
setting up, 127.
side-lever, 209.
small, 63.
trunk, 207.
vertical, how to balance, 166.
Equilibrium, conditions of, 320.
Equivalents, chemical, 334.
of inches, feet, and yards, decimal, 292.
of pounds and ounces, decimal, 292.
to the fractional parts of a gallon or an inch, decimal, 294.
Evans, Oliver, 578, 579.
Evaporation, 322.
Evaporative efficiency of a pound of anthracite coal, 337.
boilers, 372.
tubes, 412.
Everything in its place, 6.
Examination of connecting-rod, 110.
of the principles involved in the use of the crank, 111.
Examples for finding horse-power of steam-engines, 65–69.
Excessive firing, 459.
Expansion, 46.
of air by heat, table showing the, 312.
period of, 118.
Expansive property of steam, 44.
Explanation of tables of boiler pressures, 389.
working principles of condensing engine, 190.
Explosions, boiler, 452.
Explosive gases, generation of, 455.
External surfaces of tubes, table of superficial areas of, 413.

Factors, table of, 68.
Facts to be recorded when diagrams are taken, 242.
Feathering wheel, 284.
Feed-pumps for condensing engines, 183.
Feed-water heaters, 472.
Feed-water, temperature of, 176.
Film or scale, 452.
Firing, 378.
excessive, 459.
instructions for, 382.
Fitch, John, 578.
Fitting slide-valves, 103.
Fixed temperatures, 318.
Flat-head, 430.
Flue and tube cleaners, steam-boiler, 573.
boiler, marine, 398.
boilers, 352.
Flues, boiler, 417.
Fluid resistance, 287.
Fluids, elastic, 320.
Foaming, 439.
in locomotive boilers, 440.
in marine boilers, 440.
Focus, 493.
Force, 492.
centrifugal, 139.
with which steam acts, 57.
Foreign terms and units for horsepower, 59.
Form of diagrams, 250.
of cross-head, 86.
of screw, 274.

Formula by which to find positions of double eccentrics on the shaft, 107.
Friction, 493.
of slide-valves, 100.
rollers, 494.
Fuel, 46.
and air, mixture of, 335.
ingredients of, 335.
waste of unburnt, 341.
Fulton, Robert, 578.

Gang of steam-boilers, 349.
Gas, olefiant, 347.
Gases, 346.
compression and dilatation of, 348.
liquefaction of, 348.
Gauge, mercury, 231.
siphon, 231.
spring, 230.
vacuum, 226, 230.
Gauges, steam, 227, 229.
Geared engines, 207.
Gearing, 559.
Generation of explosive gases, 455.
Glass water-gauges, 233.
Good material, 72.
workmanship, 72.
Governor,
Huntoon, 140.
shaft-pulley, to find diameter of, 144.
spindles, 142.
Governors, 139.
Grate-bars, 482.
Gravity acts on gases, 320.
and gravitation, 494.
centre of, 492.
specific, 494.
Guides, 85.
strength or stiffness of, 86.
Gum, 572.
Gyration, centre of, 495.

Hampson and Whitehill's high-pressure cut-off engine, 153, 154.
Hand rock-shaft or trip-shaft, 214.
Harris Corliss high-pressure engine, 151.
Hard patch, to apply, 409.
Haskin's vertical high-pressure engine, 163, 308.
Hawkins and Dodge's high-pressure cut-off engine, 24, 147.
Heads, boiler, 429.
cross, 86.
Heat, communication of, 330.
dynamic equivalent of, 329.
latent, 326.
mechanical equivalent of, 327.
mechanical theory of, 328.
medium, 331.
molecular or atomic force of, 329.
of combustion of various fuels, table showing the total, 342.
of water or ice, latent, 301, 302.
power of expansion by, 329.
sensible, 327.
specific, 324.
table showing the expansion of air by, 312.
total or actual, 330.
transmission of, 330.
unit of, 294, 326.
upon different bodies, table showing effects of, 332.
Heater pipe, 367.
Heaters, feed-water, 472.
Heating of crank-pin, 259.
surface of steam-boilers, rule for finding the, 371.
surface to cylinder and grate surface, etc., proportions of, 402.
Heavy bearings, cooling compound for, 572.
High-pressure engine, advantages of, 54.
Corliss, 151.
desirable, 55.
economy in, 70.
Hampson and Whitehill's, 153.
Haskin's vertical, 163, 308.
Naylor's vertical, 157.
Wheelock's, 151.
High-pressure or non-condensing steam-engines, 54.
Horizontal belts, 564.
cylinders, 242.
motion, 117.

Horizontal tubes, 412.
tubular marine boiler, 404.
Horse-power, 495.
actual or net, 61.
commercial, 62.
dynamometrical, 62.
foreign terms and units for, 59.
indicated, 61.
nominal. 60.
of boilers, 375.
of condensing engines, 191.
How to adjust eccentric rod, 109.
to balance vertical engines, 166.
to calculate the power the engine is exerting when the diagram is taken, 244.
to compute a diagram, 244.
to find the throw of any eccentric, 106.
to keep indicator in order, 252.
to line up a propeller shaft, 282.
to make belts run on the centre of pulleys, 567.
to mark engineers' or machinists' tools, 569.
to put an engine in line, 128.
to put the engines in a steamboat or ship, 265.
to reverse an engine, 131.
to set a slide-valve, 132.
to test the quality of leather for belting, 567.
Huntoon governor, 140.
interior view of working parts of, 141.
Hydraulic test, 362.
Hydrocarbons, 335.
Hydrodynamics, 496.
Hydrogen, 346.
carburetted, 347.
Hydrometer, salinometer, or salt-gauge, the, 223.
Hyperbola, 496.
Hyperbolic logarithms, 517.
table of, 48, 518.

Ice, specific gravity of, 304.
Immersion of paddles, 286.
Impact, 497.
Impenetrability, 497.
Impetus, 498.
Improved methods of lacing belts, 563.
Incidence, 498.
Inclination, 498.
Inclined plane, 498, 501.
Incrustation in steam-boilers, 441.
Indicated horse-power, 61.
Indicator in order, how to keep the, 252.
method of applying, 241.
Richards', 235.
steam-engine, 235.
utility of the, 244.
Indices, the, 317.
Inertia, 498.
Ingredients of fuel, 335.
Injection and condensed water, relative quantities of, 219.
Injector, the, 168.
Rue's "Little Giant" injector, 170.
how to put on Rue's Letter "B," 170.
method of working, 171.
table of capacities of Rue's "Little Giant," 172.
Giffards' discovery of the, 169.
Injectors, instructions for setting up, 170.
table of capacities of, 172.
Inside clearance, 95.
Instructions for firing, 382.
for setting up injectors, 173.
Interior view of working parts of Huntoon governor, 141.
Internal and external corrosion of steam-boilers, 449.
radius, 470.
Invention and improvement of the steam-engine, 574.
Inverted siphon, 231.

Jackets, steam, 474.
James Watt, 187, 577.
Jet condenser, 221.
John Fitch, 578.
Joints, 571.
red-lead, 571.
Journals, centres of the, 270.

Knocking in engines, 167.

Lacing belts, improved methods of, 563.
Land and marine engines, management of, 258.
Lap on the slide-valve, 93.
required for slide-valves of stationary engines, table showing amount of, 97.
Latent caloric, 322.
heat, 326.
heat of steam, 26.
heat of various substances, 331.
heat of water or ice, 301, 302.
Laying of bed-plate, 267.
Lead of the slide-valve, 97.
Lead-wire, 572.
Leakage of air, 261.
Leather belts, 564.
Length of eccentric-rod, 109.
unit of, 294.
Leopold and Trevithick, 577.
Level the shaft, 130.
Levers, 499.
safety-valve, 436.
Lift of conical or poppet-valves, 96.
Lime, sulphate of, 443.
Linear expansion of iron, 548.
Link-motion, 116, 260.
Liquefaction of gases, 348.
"Little Giant" injector (Rue's), 170.
Load on piston-rod, 121.
Locomotive boilers, 353.
foaming in, 440.
Logarithms, 515.
from 0 to 1000, 516.
hyperbolic, 517.
Longitudinal and curvilinear strains, 387.
seams, 470.
Lord's boiler compound, 450.
Loss of pressure in cylinders induced by long steam-pipes, 475.
Lubricating steam-cylinders, 479.
thorough, 72.

Machines, 499.
Magnesia, carbonate of, 443.
Magnetic water-gauges, 234.
Magnetism, electro, 72.
Main and out-port pillow-blocks, placing of the, 270.
bearings or pillow-blocks, 114.
shaft and out-port pillow-blocks, setting of the, 269.
Management of land and marine engines, 258.
Manley paddle-wheel, 285.
Manner of placing a straight-edge, 265.
Manometer, the, 224.
Marine beam-engine, 210, 211.
boiler, horizontal tubular, 404.
flue boiler, 398.
boiler, vertical tubular, 407.
boilers, 399.
bedding, 405.
care of, 406.
clothing, 405.
corrosion of, 451.
setting, 404.
Marine-engine register, clock, and vacuum gauge, 226.
Marine-engines, eccentrics of, 106.
starting-gear for, 215.
Marine steam-engines, 197, 198.
Marquis of Worcester, 576.
Mass, 500.
Massey's rotary engine, 164, 333.
Material, good, 72.
Matter, 500.
Measurement of screw-propeller, 279.
Mechanical equivalent of heat, 327.
power, 287.
power, amount of, 288.
powers, 500.
pressure, 29.
properties of vapor, 25.
theory of heat, 328.
Mechanics, 502.
Medium heat, 351.
Mensuration of the circle, cylinder, sphere, etc., 487.
Mercurial thermometer, 313.
Mercury-gauge, 231.
Mercury, properties of, 313.
rate of expansion of, 315.
Meters, 374.

Method of applying the indicator, 241.
of determining how much power each tenant is actually using, 244.
of how to put on letter "B" injector, 170.
of working Rue's "Little Giant" injector when used to lift water, 171.
Minor causes of loss in steam-engines, 70.
Mixture of fuel and air, 335.
Modes of attaching eccentric-rods, 109.
Modulus, 502.
Molecular or atomic force of heat, 329.
Momentum, 502.
Motion, 502.
absolute, 502.
accelerated, 503.
angular, 503.
compound, 503.
horizontal, 117.
link, 116.
natural, 503.
of slide-valve, 90.
of steam, 35.
parallel, 503.
perpetual, 505.
proper points from which to derive the, 242.
relative, 503.
retarded, 503.
rotary, 104.
uniform, 503.
Movers, prime, 507.
Mud-drums, 355.
Multipliers, table of, 51.
Murdock, 579.

Natural motion, 503.
Nature of valve motion, 91.
Naylor's vertical high-pressure engine, 157, 300.
Negative slip of the screw-propeller, 274.
Neglect of steam-boilers, 362.
Net or actual horse-power, 61.
Newcomen, 577.
Nitrogen, 348.
Nominal horse-power, 60.
Non-conductor for steam-pipes and steam-cylinders, 569.

Observations on character of diagram, 243.
Ocean steamers, boilers of, 224.
Oil or tallow, 262.
Oils and oiling, 477.
Olefiant gas, 347.
Oliver Evans, 578, 579.
Ordinary radial wheel, 283.
Oscillating cylinders, 242.
engines, 205.
Oscillation, centre of, 503.
Overheating, 458.
Overpressure, 458.

Packing, piston, and valve-rod, 135.
Paddle-wheel, Manley, 285.
Paddle-wheels, 282.
disconnecting, 286.
Paddles, immersion of, 286.
Paddle-shaft, the, 286.
Parallel motions, 503.
rods, 580.
Parts of long-stroke side-valve engines, 88.
Pendulum, 504.
Percussion, 504.
Period of expansion, 118.
Perpendicular belts, 565.
Perpetual motion, 505.
Pike's boiler, tube, and flue cleaner, 573.
Pillow blocks, beam, 266.
or main bearings, 114.
Pin, rock-shaft, 86.
Pins, crank, 83.
Pirsson's surface condenser, 218.
Piston, 212, 260.
and valve-rod packing, 135.
connecting-rod, crank connection, 80.
effective pressure against, 64.
in cylinder at different crank-angles, table showing the position of, 81.

Piston packing, setting out, 134.
rings, 77.
rod, load on, 121.
rods, 83.
speeds for all classes of engines, table of, 79.
speeds, table showing proper area of steam-ports for different, 88.
springs, 77.
stroke of, 92.
Pistons, 76.
of steam-hammers, 79.
solid, 78.
steam, 78.
Pitch of a screw, 274.
Placing air-pump in its seat, 271.
of the main and out-port pillow-blocks, 270.
Plane, inclined, 498, 501.
Plates, deterioration of, 456.
Pneumatics, 506.
Points, the centre, 267.
Poppet or conical valves, 95.
Portable engines, 166.
Power, 507.
in the steam-engine, 196.
mechanical, 287, 500.
of a horse, 495.
of expansion by heat, 329.
of steam, 113.
of steam-engine, 55.
Pressure and temperature of steam, 30.
average atmospheric, 321.
of air, 310.
of steam, 28.
of steam on piston, 113.
unit of, 297.
Pressures for iron boilers, table of safe internal, 390.
steel boilers, table of safe internal, 394.
Prime movers, 507.
Priming in steam-cylinders, 476.
Propeller shaft, how to line up, 282.
Propellers, screw, 271.
Proper points from which to derive motion, 242.
proportions of screw-propellers, table of, 279.
Properties of mercury, 313.
Proportions of heating surface to cylinder and grate surface, etc., 402.
of slide-valves, 93.
of steam-cylinder, 124.
of steam-engines, 121
Pulsometer, the, 185.
Pumps, 172.
Atlas steam-pump, 179.
Dayton Cam-pump, 181.
boiler feed, 176.
steam, 178–180.
Radial wheel, 282.
Radiating power of different bodies, table of, 331.
Radiation of caloric, 322.
Radius, internal, 470.
Raising steam and getting under way, 263.
Rate of expansion of mercury, 315.
Reaumer's thermometer, 316.
Red-lead joints, 571.
Reflection of caloric, 322.
Register, marine engine, 226.
thermometers, 317.
Regulation of draft, 385.
Relative motion, 503.
positions of the beam-centre, 268.
quantities of injection and condensed water, 219.
Remedies, chemical and mechanical, 447.
Repairing steam-boilers, 409.
Resistance, fluid, 287.
of tubes, 410.
Resisting pressure, amount of, 201.
Retarded motion, 503.
Richard's indicator, 235.
Rings, brass or composition, 135.
piston, 77.
Robert Fulton, 579. (Frontispiece.)
Rock-shaft, 86.
diameter and length of, 86.
pin, 86.
the hand, 214.
Rods, eccentric, 108.
parallel, 580.
piston, 83.
valve, 85.

Rollers, friction, 494.
Roper's caloric engine, 159, 324, 325.
Rotary engine, Massey's, 333.
motion, 104.
Rubber belts, 566.
Rue's "Little Giant" injector," 170.
Rule for benefit derived from working steam expansively, 48.
Bourne's, 69.
Colburn's, 69.
for calculating horse-power of steam-engine, 68.
for calculating the number of horse-powers a belt will transmit, etc., 567.
for finding area of a circle, 489.
for finding aggregate strain, etc., on shells of steam-boilers, 389.
for finding centre of gravity of taper levers for safety-valves, 439.
for finding change required in the length of a belt, etc., 566.
for finding collapsing pressure of boiler flues, 424.
for finding horse-power of steam-engine, 65.
for finding necessary quantity of water per minute for any engine, 183.
for finding pressure, at which a safety-valve is weighted, 438.
for finding pressure per square inch, etc., 437.
for finding safe external pressure on boiler flues, 418.
for finding safe working pressure of iron boilers, 388.
for finding safe working pressure of steel boilers, 388.
for finding weight necessary to put on a safety-valve, etc., 437.
for finding the mean or average pressure in a cylinder, 50.
for finding the length of belt wanted, 566.
for finding the number of revolutions of the last wheel in a train of wheels and pinions, etc., 560.
Rule for finding the required amount of lap for slide-valve corresponding to any desired point of cut-off, 96.
for finding the required area of chimney for any boiler, 484.
for finding the required diameter of cylinder for an engine of any given horse-power, the travel of piston and available pressure being decided upon, 76.
Rules for calculating size of pulleys for governors, 144.
for comparing degrees of temperature indicated by different thermometers, 318.
for finding the diameter and speed of pulleys, 558.
for finding the diameter of toothed wheels, 559.
for finding the diameter of a pinion, etc., 559.
for finding the diameter of pump-plunger for any engine, 182.
for finding the dynamometrical or effective horse-power of a marine engine, 255.
for finding the effects of mechanical powers, 501.
for finding the heating surface of steam-boilers, 371.
for finding the number of revolutions of a pinion or driven, etc., 560.
for finding the number of revolutions of a driver, etc., 560.
for finding the number of revolutions of the last wheel at the end of a train of spur-wheels, etc., 560.
for finding the number of revolutions in each wheel for a train of spur-wheels, etc., 560.
for finding the pressure on slide-valves, 101.
for finding the quantity of water boilers, etc., are capable of containing, 386, 387.

Rules for finding the required number of teeth in a pinion, etc., 559.
  for finding the width of belt to transmit a given horse-power, 566.
Rust joints, 571.
  cement for, 568.

Safe external pressure on boiler flues, rule for finding the, 418.
  working-pressure or safe load, 470.
Safety-valve, the, 367.
  levers, 436.
Safety-valves, 431.
  table showing use of, 433.
Salinometer or salt-gauge, 223.
Salt-gauge or salinometer, 223.
Saturated steam, 32.
Screw, 501.
  as compared with the paddle, 278.
  centrifugal action of, 277.
  engine, direct-acting, 204.
  form of, 274.
  pitch of a, 274.
  propeller, measurement of, 279.
  propeller, negative slip of the, 274.
  propeller, surface of a, 279.
  propellers, 271.
  propellers, proper proportion of, table of the, 279.
Screws, twin, 278.
Seams, curvilinear, 470.
  longitudinal, 470.
Sea-water, 224.
Section of Roper's caloric engine, 325.
Self-adjusting injector, method of working the, 170.
  letter "B" injector, 170.
  adjustable injector, 171.
Sensible caloric, 322.
  heat, 327.
Set poppet or conical valves, 133.
Setting boilers, 356.
  marine boilers, 404.
  of main shaft and out-port pillow-blocks, 269.
  out piston packing, 134.
Setting up engines, 127.
  up injectors, instructions for, 171
  valves, 131.
Sewell's surface condenser, 220.
Shaft, level the, 130.
  wear of the, 73.
Shafts, crank, 114.
  rock, 86.
Side-lever engines, 209.
  pipes, the, 213.
Signification of signs used in calculations, 291.
Signs used in calculations, signification of, 291.
Single- and double-riveted seams, comparative strength of, 461.
Siphon, inverted, 231.
Siphon-gauge, 231.
Size of pulleys for governors, rules for calculating the, 144.
Slides for the cross-head, 269.
Slide-valve, 580.
  common, 94.
  connections of, 103.
  corresponding to any desired point of cut-off, rule for finding the required amount of lap for a, 96.
  engines, ports of long stroke, 88.
  how to set a, 132.
  lap on the, 93.
  lead of the, 97.
  motion of, 90.
Slide-valves, 89.
  balanced, 102.
  fitting, 103.
  friction of, 100.
  proportions of, 93.
  rule for finding the pressure on, 101.
Slip of the screw-propeller, negative, 274.
Small engines, 63.
Smoke, 485.
Soapstone, 136.
Soft patch, the, 410.
Solder, 570.
Solid pistons, 78.
  thermometers, 318.
Solvents, 448.

Sound test, 362.
Specific gravity, 494.
gravity of ice, 304.
gravity of water, 304.
heat, 324.
Spheroidal theory, 456.
Spindles, governor, 142.
Spirit thermometers, 318.
Spontaneous combustion, 342.
Spring-gauge, the, 230, 581.
Springs, piston, 77.
Starting-gear for marine engines, 215.
Statics, 507.
Stay-bolts, 468.
Steam, 25.
atmospheric pressure of, 32.
at different pressures, table showing temperature and weight of, 39–43.
effluent velocity of, 32.
elasticity of, 27.
expansive property of, 44.
latent or concealed heat of, 26.
mechanical pressure of, 29.
motion of, 35.
on piston, pressure of, 113.
power of, 113.
pressure of, 28.
saturated, 32.
superheated, 34.
temperature and pressure of, 30.
upon piston, table showing average pressure of, 45, 50.
volume and weight of, 35.
water while passing into, 29.
Steam-boiler, flue, and tube cleaners, 573.
Steam-boilers, 350.
care and management of, 363.
gang of, 349.
incrustation in, 441.
internal and external corrosion of, 449.
neglect of, 362.
repairing, 409.
Steam-chest, 85, 213.
Steam-cylinder, proportions of, 124.
thickness of, 122.
Steam-cylinders, lubricating, 479.
Steam-cylinders of different diameters table showing proper thickness for, 75.
priming in, 476.
thickness of, 75.
Steam domes, 353.
Steam-engine, 21.
economical working of, 73.
indicator, 235.
invention and inprovement of the, 574.
minor causes of loss in, 70.
power in the, 196.
power of, 55.
rule for calculating horse-power of, 68.
theory of the, 298.
valuable features of, 73.
waste in, 70.
Steam-engines, condensing or low-pressure, 188.
design of, 73.
high-pressure or non-condensing, 54.
marine, 197, 198.
proportions of, 121.
rule for finding horse-power of, 65.
variety of designs of, 74.
Steam-gauges, 227, 229.
hammers, pistons of, 79.
jackets, 474.
joints and patching steam-boilers, cement for making, 568.
lever, 214.
pipes and steam-cylinders, non-conductors for, 569.
pistons, 78.
Steam-ports, dimensions of, 87.
value, 87.
Steam-pumps, 180.
directions for setting up, 184.
Steam-room, 401.
Steeple-engine, the, 208.
Straight-edge, manner of placing, 265.
Strains, longitudinal and curvilinear, 387.
Strength of stayed and flat surfaces, 468.

Strength or stiffness of guides, 86.
Stroke and number of revolutions for different piston speeds in feet per minute, table showing length of, 82.
of piston, 92.
Subdivision of crank-circle, 112.
Substances, latent heat of, 331.
Sulphate of lime, 443.
Superheated steam, 34.
Surface condenser, Pirsson's, 218.
Sewell's, 220.
unit of, 295.

Table containing diameters, circumferences, and areas of circles, etc., 511.
containing diameters, circumferences, and areas of circles from $\frac{1}{16}$ of an inch to 100 inches, etc., 521.
deducted from experiments on iron plates for steam-boilers, etc., 470.
of capacities of injectors, 177.
of coefficients of frictions between plane surfaces, 480.
of comparison between experimental results and theoretical formulæ, 435.
of factors, 68.
of hyperbolic logarithms, 48, 518.
of multipliers, 51.
of piston speeds for all classes of engines, 79.
of proper proportions of screw-propellers, 279.
of radiating power of different bodies, 331.
of safe internal pressures for iron boilers, 390.
of safe internal pressures for steel boilers, 394.
of safe working external pressures on flues ten feet long, 420.
of safe working external pressures on flues twenty feet long, 422.
of squares, cubes, and square and cube roots, etc., 532.
Table of squares of thicknesses of iron, etc., 419.
of superficial areas of external surfaces of tubes, etc., 413.
of temperatures, etc., 345.
showing actual extension of wrought-iron at various temperatures, 548.
showing amount of lap required for slide-valves of stationary engines, etc., 97.
showing angular position of crank-pin, 84.
showing average pressure of steam upon piston, 49, 50.
showing boiling-point for fresh water at different altitudes above sea-level, 306.
showing effects of heat upon different bodies, 332.
showing length of stroke and number of revolutions for different piston speeds, etc., 82.
showing nature and value of varieties of American coal and coke, etc., 343.
showing position of piston in cylinder at different crank-angles, 81.
showing prominent qualities in principal American woods, 344.
showing proper area of steam-ports for different piston speeds, 88.
showing proper diameter and height of chimney for any kind of fuel, 484.
showing proper thickness for steam-cylinders of different diameters, 75.
showing relative properties of good coke, coal, and wood, 344.
showing result of experiments made on different brands of boiler-iron, etc., 471.
showing rise of safety-valves, etc., at different pressures, 433.
showing temperature and weight of steam at different pressures, 39–43.

Table showing tensile strength of various qualities of cast-iron plates, 549.
showing tensile strength of various qualities of steel plates, 549.
showing tensile strength of various qualities of wrought-iron, 547.
showing the expansion of air by heat, etc., 312.
showing the units of heat, etc., 473.
showing total heat of combustion of various fuels, 342.
showing weight of atmosphere in pounds. etc., 311.
showing weight of boiler-plates one foot square, etc., 551.
showing weight of cast-iron balls etc., 553.
showing weight of cast-iron pipes one foot in length, etc., 554.
showing weight of different materials, 571.
showing weight of round bar-iron, etc., 552.
showing weight of square bar-iron, 551.
showing weight of water, 305.
showing weight of water at different temperatures, 305.
showing weight of water in pipe, etc., 307.
showing weight per square foot of wrought-iron, steel, copper, and brass, 555.
Tables of collapsing pressure of wrought-iron boiler-flues, etc., 425, 428.
Tallow or oil, 462.
Temperature and pressure of steam, 30.
of feed-water, 176, 177.
Temperatures, fixed, 318.
underground, 318.
Tendency to overheating, 446.
Tensile strength, 469.
Test, hydraulic, 362.
sound, 362.
Testing boilers, 359.
Theoretical curve, 238.
Theory of the steam-engine, 298.
Thermometer, 313.
centigrade, 315.
mercurial, 313.
Reaumer's, 316.
Thermometers, comparative scale of Centigrade, Fahrenheit, and Reaumer, 314.
register, 317.
spirit, 318.
solid, 318.
Thickness of steam-cylinders, 75, 122.
Thorough lubrication, 72.
Throw or stroke of the eccentric, 106.
Tighteners, 567.
Time or duration, unit of, 296.
To find diameter of engine-shaft pulley, 144.
find diameter of governor shaft-pulley, 144.
polish brass, 570.
set poppet or conical valves, 133.
Tools, 508.
Torsion, 508.
Total or actual heat, 330.
Transmission of heat, 330.
Trevithick and Leopold, 577.
Trip-shaft or hand rock-shaft, the 214.
Trunk-engines, 207.
Tubes, crowding, 411.
diameter and arrangement of, 411.
evaporative efficiency of, 412.
horizontal, 412.
resistance of, 410.
use of, 410.
Tubular boiler, 352.
Twin-screws, 278.

Underground temperatures, 318.
Uniform motion, 503.
Unit of capacity, 295.
of heat, 294.
of length, 294.
of pressure, 297.
of surface, 295.
of time or duration, 296.
of velocity, 297.
of weight, 296.

Unit of work, 297.
Units, 294.
Use of tubes, 410.
Useful numbers in calculating weights and measures, etc., 292.
Utility of indicator, 244.

Vacuum, 193.
gauge, 226, 230.
Valuable features of steam-engine, 73.
Value of steam-ports, 87.
Valve-gear, 213.
motion, nature of, 91.
rods, 85.
slide, 580.
Valves, 213.
Corliss, 125.
lift of conical or poppet, 96.
poppet, or conical, 95.
safety, 431.
setting the, 131.
to set poppet or conical, 133.
Vapor, mechanical properties of, 25.
Variety of designs of steam-engines, 74.
Velocity, 508.
unit of. 297.
Vertical compound engine, 199.
cylinders. 241.
engines, how to balance, 166.
tubular marine boiler, 407.
Volume and weight of steam 35.

Waste in the steam-engine, 70.
of unburnt fuel, 31.
Water, 299.
boiling-point of, 303.
composition of. 299.
Water-gauges, glass, 233.
magnetic, 234.
Water, necessary quantity of, rule for finding. 133.
passing into steam, 29.
specific gravity of, 304.
weight of, table showing, 305.
Watt, James, 187, 577.
Watt and Campbell's high-pressure cut-off engine, 37, 481.
Watt's rule for calculating power of condensing engines, 191.
Wear of the shaft, 73.
Wedge, 501.
Weight, 509.
and volume of steam, 35.
of atmosphere in pounds, etc., table showing the, 311.
of cast-iron plates per superficial foot as per thickness, 553.
of shafting, etc., calculating the, 552.
of water at different temperatures, table showing the, 305.
of water in pipe, etc., table showing the, 307.
unit of, 296.
Weights and measures, 509.
Wheel and axle, 501.
feathering, 284.
ordinary radial, 283.
radial, 282.
Wheelock's high-pressure cut-off engine, 71, 151.
Wheels, cycloidal, 283.
fly, 115.
paddle, 282.
Williams' three-cylinder engine, 319.
vertical three-cylinder high-pressure engine, 158, 319.
Woodruff and Beach's high-pressure cut-off engine, 156, 165.
Worcester, Marquis of, 576.
Work, 510.
unit of, 297.
Working-beam, 211.
Working steam expansively, 43.
strength, 469.
Workmanday, 510.
Workmanship, good, 72.
Wright's high-pressure cut-off engine, 20, 146.
Wrought-iron at various temperatures, table showing actual extension of, 548.
linear expansion of, 548.
various qualities of, table showing tensile strength of, 547.

Zero, absolute, 316.
change of, 316.

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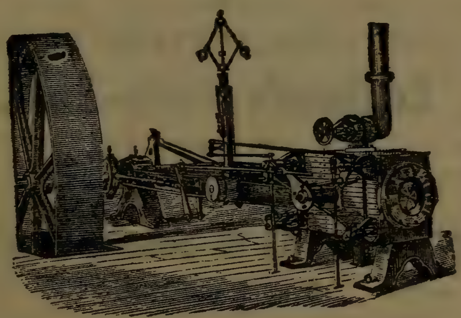

Of all the efforts of human ingenuity known, perhaps none has monopolized so large a share of inventive genius as the steam-engine. No other object in the entire range of human devices has so irresistibly arrogated to itself the devotion of scientific men as the production of an artificial movement from the vapor of boiling water.

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INTRODUCTION.

THE object of the writer in preparing these works has been to present to the practical engineer a set of books to which he can refer with confidence for information regarding every branch of his profession. Up to the date of the publication of these books, it was impossible to find a plain and practical treatise on the steam-engine. This arose, perhaps, from the fact that men who had attained proficiency in steam-engineering had no taste for devoting their limited leisure time to writing, and that those whose circumstances enabled them to do so, were precluded from a want of that practical knowledge which is only obtained by years of hard work and close observation. Many of the books heretofore written on the steam-engine are full of formulæ for calculating questions that may arise in the engine-room; but, as they are generally expressed in algebraical form, they are of little service to the majority of engineers; for, however useful such formulæ may be to the scientific, they can be of no practical value to men who do not fully understand them. It is also no less a fact, that nearly all writers on the steam-engine deal more with the past than the present. This is to be regretted, for, however interesting the bygone records of steam-engineering may be as a history, they cannot instruct the engineer of the present day in the principles and practice of his profession.

An experience of over thirty years, with all kinds of

engines and boilers, enables the writer to fully understand the kind of information most needed by men having charge of steam-engines of every description, and what they could comprehend and employ. With this object in view, he has carefully investigated all the details of stationary, locomotive, fire, and marine engines, taking up each subject singly, and excluding therefrom everything not directly connected with steam-engineering. Particular attention has been given to the latest improvements in all these classes of engines, and their proportioning according to the best modern practice, which will be of immense value to engineers, as nothing of the kind has heretofore been published. They also contain ample instructions for setting up, lining, reversing, and setting the valves of all classes of engines—subjects that have not received that attention from other writers on the steam-engine which their importance so justly merits. A certain portion of each book is devoted to an examination and discussion of the principles of Hydro- and Thermo-Dynamics, which include Air, Water, Heat, Combustion, Steam, Liquefaction, Dilatation of Gases, Molecular and Atomic Forces, Dynamic Equivalents, subjects with which the practical engineer should be fully conversant; as to ignore the principles of any subject is similar to building a structure without knowing the strength of the foundation; for it is only by a minute and careful analysis of the physical phenomena which convert heat into a motor force that the steam-engine has been brought to its present perfection.

S. R.

HAND-BOOK

OF

LAND AND MARINE ENGINES;

INCLUDING

THE MODELLING, CONSTRUCTION, RUNNING, AND MANAGEMENT OF LAND AND MARINE ENGINES AND BOILERS.

Fully Illustrated.

BY

STEPHEN ROPER, ENGINEER,

Author of

"Roper's Catechism of High-Pressure or Non-Condensing Steam-Engines," "Roper's Hand-Book of the Locomotive," "Roper's Hand-Book of Modern Steam Fire-Engines," "Roper's Handy-Book for Engineers," "Roper's Improvements in Steam-Engines," "Roper's Use and Abuse of the Steam-Boiler," "Questions for Engineers," etc.

MARINE BEAM-ENGINE.

PHILADELPHIA:

E. CLAXTON & COMPANY.

Roper's Hand-book of Land and Marine Engines.

Opinions of the Press.

Iron Age, New York.

THE author of this hand-book says, in his preface, that his object in preparing it, "has been to present to the practical inquirer a book to which he can refer with confidence for information in regard to every branch of his profession."

Rules and directions expressed in algebraic formulæ are of little service to the majority of engineers, because they are not fully understood. The author, keeping this in mind, has avoided most of the points which render many of our hand-books of limited value to the practical man. He has had a long and extensive practical experience among the men for whom he writes, and understanding their wants, has produced a book which seems admirably adapted to those who have anything to do, in a practical way, with steam machinery. We have given the work a careful examination, and consider it one of the most satisfactory works of the kind we have ever seen. Mr. Roper thoroughly understands his subject, being entirely practical, and, at the same time, having a correct understanding of scientific principles. His chapters on the theory of steam engineering are so simple and practical that there is no mechanic in the country, however ignorant he may be of higher mathematics, who cannot learn all they are intended to teach. His practical directions for the management of engines are just such as we should expect from an experienced engineer who had spent all his

life in an engine-room, but who had learned the theory as well as the practice of his trade. They are plain and to the point, and the reader may accept them with an entire confidence. His descriptions of engines, pumps, and the appliances connected with engines, are exceedingly satisfactory, as are also his rules, which seem to be the best and simplest which could be formulated. The book has an abundance of tabular information, which seems to include all the tables that could be of any use. The engravings are good, and are just what is wanted to explain the text. In a word, the amount and kind of information contained in this work seems to be all that could be desired. The owner of a steam-engine cannot well do without it, and no one who runs an engine should be ignorant of any part of its contents.

Scientific American, New York.

THE house of Messrs. E. Claxton & Company have just published a new work on "Land and Marine Engines," by Stephen Roper, Engineer, author of "Roper's Catechism of High-Pressure or Non-Condensing Steam-Engines," and "Roper's Hand-Book of the Locomotive," etc. Mr. Roper needs no introduction to our readers as a competent and trustworthy authority on steam engineering, and his present volume will prove exceedingly valuable to all engineers who desire a treatise combining scientific accuracy with a popular style — free from formulæ and ultra-mathematical expressions. The Tables with which the work is interspersed are numerous and valuable, and there is at the end of the book a very interesting historical account of the invention and improvement of the steam-engine.

CONTENTS.

INTRODUCTION.
THE STEAM-ENGINE.
STEAM.
Table showing the Temperature and Weight of Steam at different Pressures from one Pound per Square Inch to 300 Pounds, and the Quantity of Steam produced from 1 Cubic Inch of Water, according to Pressure.
ECONOMY OF WORKING STEAM EXPANSIVELY.
Table of Hyperbolic Logarithms to be used in Connection with the above Rule.
Table showing the average Pressure of Steam upon the Piston throughout the Stroke, when Cut-off in the Cylinder from $\frac{1}{3}$ to $\frac{7}{11}$, commencing with 25 Pounds and advancing in 5 Pounds up to 130 Pounds Pressure.
Table of Multipliers by which to find the mean Pressure of Steam at various points of Cut-off.
HIGH-PRESSURE OR NON-CONDENSING STEAM-ENGINES.
POWER OF THE STEAM-ENGINE.
Foreign Terms and Units for Horse-power.
Table of Factors.
WASTE IN THE STEAM-ENGINE.
DESIGN OF STEAM-ENGINES.
THE BED-PLATE.
CYLINDERS.
Table showing the proper Thickness for Steam-cylinders of different Diameters.

CONTENTS.

PISTONS.
PISTON-RINGS.
PISTON-SPRINGS.
STEAM-PISTONS.
SOLID PISTONS.
Table of Piston Speeds for all Classes of Engines — Stationary, Locomotive, and Marine.
PISTON, CONNECTING-ROD, AND CRANK CONNECTION.
Table showing the Position of the Piston in the Cylinder at different Crank-angles, according to the length of Connecting-rod.
Table showing length of Stroke and Number of Revolutions for different Piston Speeds in Feet per Minute.
PISTON-RODS.
CRANK-PINS.
Table showing the Angular Position of the Crank-pin corresponding with the various Points in the Stroke which the Piston may occupy in the Cylinder.
STEAM-CHESTS.
VALVE-RODS.
GUIDES.
ROCK-SHAFTS.
CROSS-HEADS.
STEAM-PORTS.
Table showing the Proper Area of Steam-ports for different Piston Speeds.
SLIDE-VALVES.
PROPORTIONS OF SLIDE-VALVES.
LAP ON THE SLIDE-VALVE.
POPPET OR CONICAL VALVES.
Table showing the Amount of "Lap" required for Slide-valves of Stationary Engines when the Steam is to be Worked expansively.

CONTENTS.

Lead of the Slide-valve.
Clearance.
Compression.
Friction of Slide-valves.
Balanced Slide-valves.
Fitting Slide-valves.
Slide-valve Connections.
Eccentrics.
Eccentric-rods.
Cranks.
Crank-shafts.
Pillow-blocks, or Main Bearings.
Fly-wheels.
Link-motion.
Proportions of Steam-engines according to the best Modern Practice.
Setting up Engines.
Dead-centre.
How to put an Engine in Line.
How to Reverse an Engine.
Setting Valves.
How to set a Slide-valve.
Setting out Piston Packing.
Piston- and Valve-rod Packing.
Automatic Cut-offs.
Governors.
The Huntoon Governor.
The Allen Governor.
The Cataract.
Wright's High-pressure Engine.
Hawkins and Dodge's High-pressure Engine.
Watts and Campbell's High-pressure Engine.
The Buckeye High-pressure Engine.

Wheelock's High-pressure Engine.
The Corliss High-pressure Engine.
Hampson and Whitehill's High-pressure Engine.
The Allen High-pressure Engine.
Woodruff and Beach's High-pressure Engine.
Naylor's Vertical High-pressure Engine.
Williams' Vertical Three-cylinder High-pressure Engine.
Roper's Caloric Engine.
Haskins' Vertical High-pressure Engine.
Massey's Rotary Engine.
Portable Engines.
How to Balance Vertical Engines.
Knocking in Engines.
The Injector.
Pumps.
Force-pumps.
Piston-pumps.
Boiler Feed-pumps.
Steam-pumps
The Atlas Steam-pump.
The Dayton Cam-pump.
Directions for setting up Steam-pumps.
The Pulsometer.
James Watt.
Condensing or Low-pressure Steam-engines.
Explanation of the working Principles of the Condensing Engine.
Horse-power of Condensing Engines.
The Vacuum.
Marine Steam-engines.
Compound Engines.

Direct-acting Engines.
Balancing the Momentum of Direct-acting Engines.
Oscillating Engines.
Trunk Engines.
Geared Engines.
Back-action Engines.
Side-lever Engines.
Beam Engines.
Marine Beam Engine.
Starting-gear for Marine Engines.
Condensers.
Air-pumps.
The Hydrometer, Salinometer, or Salt-gauge.
The Manometer.
The Barometer.
Marine Engine Register, Clock, and Vacuum Gauge.
Steam-gauges.
Glass Water-gauges.
The Steam-engine Indicator.
Method of Applying the Indicator.
Form of Diagrams.
How to keep the Indicator in Order.
The Dynamometer.
The Engineer.
Management of Land and Marine Engines.
How to put the Engines in a Steamboat or Ship.
Screw-propellers.
Paddle-wheels.
Fluid Resistance.
Signification of Signs used in Calculations.

CONTENTS.

DECIMAL.
Decimal Equivalents of Inches, Feet, and Yards.
Decimal Equivalents of Pounds and Ounces.
Useful Numbers in calculating Weights and Measures, etc.
Decimal Equivalents to the Fractional Parts of a Gallon or an Inch.
UNITS.
THEORY OF THE STEAM-ENGINE.
WATER.
AIR.
THE THERMOMETER.
Comparative Scale of Centigrade, Fahrenheit, and Reaumer Thermometers.
ELASTIC FLUIDS.
CALORIC.
HEAT.
COMBUSTION.
GASES.
STEAM-BOILERS.
STEAM-DOMES.
MUD-DRUMS.
SETTING BOILERS.
EXPANSION AND CONTRACTION OF BOILERS.
TESTING BOILERS.
NEGLECT OF STEAM-BOILERS.
CARE AND MANAGEMENT OF STEAM-BOILERS.
HEATING SURFACE.
RULES FOR FINDING THE HEATING SURFACE OF STEAM-BOILERS.
EVAPORATIVE EFFICIENCY OF BOILERS.
HORSE-POWER OF BOILERS.

FIRING.
INSTRUCTIONS FOR FIRING.
RULES FOR FINDING THE QUANTITY OF WATER BOILERS AND OTHER CYLINDRICAL VESSELS ARE CAPABLE OF CONTAINING.
LONGITUDINAL AND CURVILINEAR STRAINS.
RULES.
EXPLANATION OF TABLES OF BOILER PRESSURES ON FOLLOWING PAGES.
Table of safe Internal Pressures for Iron Boilers.
Table of safe Internal Pressures for Steel Boilers.
MARINE BOILERS.
Proportions of Heating Surface to Cylinder and Grate Surface of noted Ocean, River, and Ferry-boat Steamers.
SETTING MARINE BOILERS.
BEDDING MARINE BOILERS.
CLOTHING MARINE BOILERS.
CARE OF MARINE BOILERS.
REPAIRING STEAM-BOILERS.
TUBES.
Table of Superficial Areas of External Surfaces of Tubes of Various Lengths and Diameters in Square Feet.
BOILER FLUES.
BOILER-HEADS.
SAFETY-VALVES.
Table showing the Rise of Safety-valves, in Parts of an Inch, at different Pressures.
RULES.
FOAMING.
INCRUSTATION IN STEAM-BOILERS.

CONTENTS.

INTERNAL AND EXTERNAL CORROSION OF STEAM-BOILERS.
BOILER EXPLOSIONS.
COMPARATIVE STRENGTH OF SINGLE AND DOUBLE-RIVETED SEAMS.
CALKING.
STRENGTH OF THE STAYED AND FLAT SURFACES.
DEFINITIONS AS APPLIED TO BOILERS AND BOILER MATERIALS.
FEED-WATER HEATERS.
Table showing the Units of Heat required to Convert One Pound of Water, at the Temperature of 32° Fah., into Steam at different Pressures.
STEAM-JACKETS.
LOSS OF PRESSURE IN CYLINDERS INDUCED BY LONG STEAM-PIPES.
PRIMING IN STEAM-CYLINDERS.
OILS AND OILING.
Table of Coefficients of Frictions between Plane Surfaces.
GRATE-BARS.
CHIMNEYS.
Table showing the proper Diameter and Height of Chimney for any kind of Fuel.
SMOKE.
MENSURATION OF THE CIRCLE, CYLINDER, SPHERE, ETC.
CENTRAL AND MECHANICAL FORCES AND DEFINITIONS.
THE CIRCLE.
Table containing the Diameters, Circumferences, and Areas of Circles, and the Contents of each in Gallons, at 1 Foot in Depth.

CONTENTS.

LOGARITHMS.
Table of Logarithms of Numbers from 0 to 1000.
HYPERBOLIC LOGARITHMS.
Table of Hyperbolic Logarithms.
Table containing the Diameters, Circumferences, and Areas of Circles from $\frac{1}{16}$ of an Inch to 100 Inches.
RULES FOR FINDING THE DIAMETER AND SPEED OF PULLEYS.
GEARING.
BELTING.
CEMENT FOR MAKING STEAM-JOINTS AND PATCHING STEAM-BOILERS.
NON-CONDUCTORS FOR STEAM-PIPES AND STEAM-CYLINDERS.
HOW TO MARK ENGINEERS' OR MACHINISTS' TOOLS.
TO POLISH BRASS.
SOLDER.
Table showing Weight of different Materials.
JOINTS.
THE INVENTION AND IMPROVEMENT OF THE STEAM-ENGINE.

HAND-BOOK

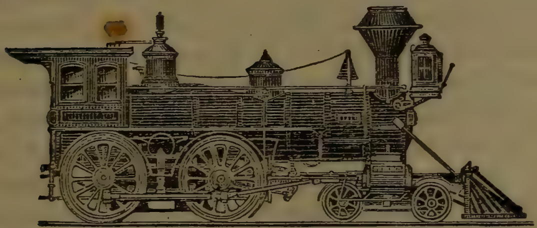
OF THE

LOCOMOTIVE;

INCLUDING THE

CONSTRUCTION, RUNNING, AND MANAGEMENT
OF LOCOMOTIVE ENGINES AND BOILERS.

Fully Illustrated.



BY

STEPHEN ROPER, ENGINEER,

Author of
"Roper's Catechism of High-Pressure or Non-Condensing Steam-Engines," "Roper's Hand-Book of Land and Marine Engines," "Roper's Hand-Book of Modern Steam Fire-Engines," "Roper's Handy-Book for Engineers," "Roper's Improvements in Steam-Engines," "Roper's Use and Abuse of the Steam-Boiler," "Questions for Engineers," etc.

PHILADELPHIA:
E. CLAXTON & COMPANY.

ROPER'S HAND-BOOK
OF
THE LOCOMOTIVE.

OPINIONS OF THE PRESS.

Scientific American, New York.

The author of this work very truly believes that in a book, as in a clock, any complication of its machinery has a tendency to impair its usefulness and affect its reliability. Hence, in preparing a book which is intended to be a guide for the practical locomotive engineer, he avoids "mathematical problems and entangling formulæ," and offers a pocket volume, full of information, theoretical as well as practical, succinctly and clearly condensed. There are chapters on heat, combustion, water, air, gases and steam; others on the construction of the locomotive and of its various parts, entered into with considerable details; instructions for the care and management of boilers and engines, tables of strength of materials, and useful practical hints for the guidance of the engineer. In brief, the volume is, as its name indicates, a hand-book to which the locomotive mechanic can turn for information regarding almost every branch of his trade. It is neatly illustrated and bound in morocco, in convenient pocket-book form.

North American and United States Gazette, Phila.

Mr. Roper asserts as a preliminary qualification for his task, that he has had more than thirty years' experience with all

classes of steam-engines and boilers. The object of the work is to convey practical knowledge of all that appertains to the locomotive engine and boiler, in a practical manner. Stationary and marine engines are omitted, because other treatises furnish all that need be known of them. Mr. Roper seems to know *exactly* what the class for whom he writes require, and what they can comprehend and employ. His opinion, as expressed in his work, is the highest compliment ever paid to those in question, and to the railways of this country, by which this skill has been created and is sustained and promoted. The mechanical and dynamical equivalents of heat and its molecular force are treated in a clear and lucid manner. Chemical equivalents, the liquefaction and dilatation of gases, superheated steam, tractive and evaporative power, combustion, mensuration, incrustation, and similar subjects are discussed. The strictly mechanical information is fully and lucidly set forth, to an extent that would gain a degree in any of our schools. But beyond the rudiments, and beyond their combinations and applications, there is the pervading idea that the American engineer aims to know the effect by its cause—seeks philosophical knowledge as a part of his employment, and not only seeks, but, as a whole, has mastered so much that he deserves a standard in pure science very few have supposed. No higher compliment could be paid, and it could be paid nowhere else. The treatise apparently omits nothing, expresses clearly though compactly, furnishes tables, and is a fine tribute to the practical ability of the country. It contains suitable illustrations, and is appropriately prefaced with a portrait of M. W. Baldwin.

CONTENTS.

INTRODUCTION.
THE LOCOMOTIVE.
LOCOMOTIVE ENGINEERS.
THEORY OF THE LOCOMOTIVE.
WATER.
AIR.
COMPARATIVE SCALE OF ENGLISH, FRENCH, AND GERMAN THERMOMETERS.
THE THERMOMETER.
ELASTIC FLUIDS AND VAPORS.
CALORIC.
HEAT.
COMBUSTION.
GASES.
STEAM.
Table showing the Velocity with which Steam of Different Pressures will flow into the Atmosphere or into Steam of lower Pressure.
Rule for finding the Superficial Feet of Steam-pipe required to Heat any Building with Steam.
Table showing the Temperature of Steam at Different Pressures from 1 pound per Square Inch to 240 pounds, and the Quantity of Steam produced from a Cubic Inch of Water, according to Pressure.
HORSE-POWER OF STEAM-ENGINES.
Rule for finding the Horse-power of Stationary Engines.
THE POWER OF THE LOCOMOTIVE.

Rule for finding the Horse-power of a Locomotive.
Rules for calculating the Tractive Power of Locomotives.
Table of Gradients.
Adhesive Power of Locomotives.
Proportions of Locomotives, according to best Modern Practice.
Proportions of Different Parts of Locomotives, according to best Modern Practice.
Table showing the Travel of Valve and the Amount of Lap and Lead for Different Points of Cut-off, and the Distance the Steam follows the Piston on the Forward Motion.
RULES.
LOCOMOTIVE BUILDING.
CONSTRUCTION OF LOCOMOTIVES.
SETTING THE VALVES OF LOCOMOTIVES.
DEAD WEIGHT IN LOCOMOTIVES.
Table showing the number of Revolutions per minute made by Drivers of Locomotives of different Diameters and at different Speeds.
STEAM-PORTS.
BRIDGES.
ECCENTRICS.
ECCENTRIC RODS.
Formula by which to find the Positions of the Eccentric on the Shaft.
THE SLIDE-VALVE.
FRICTION ON THE SLIDE-VALVE.
LAP AND LEAD OF VALVE.
BALANCED SLIDE-VALVE.
Table showing the Amount of Lap and Lead on the

Valves of Locomotives in Practice, on thirty-five of the principal Railroads in this Country.
The Link.
Adjustment of the Link.
Steam and Spring Cylinder Packing for Locomotives.
Rule for finding the size of Piston- and Valve-rod Packing.
Brasses for Driving-axles of Locomotives.
Lateral Motion.
Speed Indicators.
Locomotive Boilers.
Proportions of the Locomotive Boiler, from the best Modern Practice.
Wagon-top and Straight Boilers.
The evaporative Power of Locomotive Boilers.
Heating Surface, Steam Room, and Water Space in Locomotive Boilers.
Heating Surface to Grate Surface in Steam Boilers.
Rule for finding the Heating Surface in Locomotive Boilers.
Rule for finding the Heating Surface in the Tubes of Locomotive Boilers.
Rule for finding the Heating Surface in Stationary Boilers.
Punched and Drilled Holes for the Seams of Locomotive Boilers.
Machine and Hand Riveting for Locomotive Boilers.
Comparative Strength of Single and Double Riveted Boiler Seams.

Furnaces of Locomotive Boilers.
Proportions of Fire-boxes, from the best Modern Practice.
Strength of Stayed Surfaces in the Furnaces of Locomotive Boilers.
Stay-bolts.
Crown-bars.
Tubes.
Combustion of Fuel in Locomotive Furnaces.
Smoke-box.
Smoke-stacks.
Exhaust-nozzle.
Safety-valves.
Table showing the Rise of the Safety-valves.
Steam-gauges.
Instructions for the Care and Management of Locomotive Boilers.
Firemen on Locomotives.
Firing.
The Injector.
Signals.
Wrecking Tools.
Rules for finding the Elasticity of Steel Springs.
Central and Mechanical Forces and Definitions.
Table containing Diameters, Circumferences, and Areas of Circles, etc.
Incrustation in Steam-boilers.
Boiler Explosions.
Vocabulary of Technical Terms as applied to the different parts of Locomotives.

HAND-BOOK

OF MODERN

STEAM FIRE-ENGINES;

INCLUDING

THE RUNNING, CARE, AND MANAGEMENT OF STEAM FIRE-ENGINES AND FIRE-PUMPS.

With Illustrations.

BY

STEPHEN ROPER, Engineer,

Author of
"Roper's Catechism of High-Pressure or Non-Condensing Steam-Engines," "Roper's Hand-Book of the Locomotive," "Roper's Hand-Book of Land and Marine Engines," "Roper's Engineer's Handy-Book," "Roper's Improvements in Steam-Engines," "Roper's Use and Abuse of the Steam-Boiler," "Questions for Engineers," etc., etc.

PHILADELPHIA:
E. CLAXTON & COMPANY.

24

ROPER'S HAND-BOOK

OF

MODERN STEAM FIRE-ENGINES.

OPINIONS OF THE PRESS.

The Iron Age, N. Y.

IN this work the author has done for the steam fire-engine what has been long needed, by giving us a manual which is as useful as any of those which treat of the steam-engine in other and better understood forms. This, if we are not mistaken, has never been done before, and what a fireman needed to know about his "steamer" he had to learn by experience or word of mouth, or else go in ignorance. Mr. Roper has aimed to tell just what a man needs to know about the care and management of this class of engines. The work was a difficult one, but the author has been very successful. The attempt to make literature, or begin a literature for a new department of mechanics, is a very difficult one, and this latter is the task which the author has been called upon to undertake. It is comparatively easy to compile a text-book or pocket manual when the matter is already at hand, but in this case there has been very little available matter, the greater part of the work apparently being original and gathered from a great variety of sources. The tables of performances are very interesting, and are the first which we remember to have seen. Proportions for pumps of various kinds is another valuable table. Much that pertains to the machine simply as a steam-engine is, of course, similar to what would be found in any treatise upon the steam-engine. The work is a very creditable one to the author, and is, in some respects, an improvement upon any of his earlier works.

CONTENTS.

The Steam Fire-engine.
Fire.
Precautions against Fires.
What to do in Case of Fire.
Means of Preventing Fires.
Different Methods of Extinguishing Fires.
Fire-escapes.
Fire-proof Buildings.
Losses by Fire.
Ahrens' Steam Fire-engine.
Air.
Elastic Fluids.
Air-vessels.
Clapp and Jones' Steam Fire-engine.
Water.
Silsby Rotary Steam Fire-engine.
Method of Working the Steam in the Silsby Rotary Engine.
Rules.
Steam Fire-engines.
Names of Principal Manufacturers of Steam Fire-engines in this Country.
Amoskeag Steam Fire-engine.
Early Forms of Steam Fire-engines.
Floating Steam Fire-engines.
Self-propelling Steam Fire-engines.

CONTENTS.

The Button Steam Fire-engine.
Trials of Steam Fire-engines.
Instructions for the Care and Management of Steam Fire-engines and Boilers.
Engineers.
Firemen.
Useful Information for Engineers and Firemen.
Paid and Volunteer Fire Departments.
Fire-alarms.
The Gould Steam Fire-engine.
Routine of Business in Paid Fire Departments.
The Jeffer's Steam Fire-engine.
Fire-hose.
Hose-couplings.
Dimensions of First and Second Class Steam Fire-engines.
Horizontal Distances thrown by Modern Steam Fire-engines.
Perpendicular Heights thrown by Modern Steam Fire-engines.
High-pressure or Non-condensing Steam-engines — Fire, Locomotive, and Stationary.
Power of the Steam-engine.
Foreign Terms and Units for Horse-power.
The Power or Horse-power of the Locomotive.
Rules for Calculating the Tractive Power of Locomotives.
Neafie and Levy's Steam Fire-engine.
Waste in the High-pressure or Non-condensing Steam-engines.
The Ives' Steam Fire-engine.
Different Parts of Steam-engines — The Crank.

Table showing the Angular Position of the Crank-pin Corresponding with the various Points in the Stroke which the Piston may occupy in the Cylinder.
Table of Piston Speeds for all Classes of Engines—Stationary, Locomotive, Fire, and Marine.
Table showing Position of the Piston in the Cylinder at different Crank-angles, according to the length of Connecting-rod.
Table showing Length of Stroke and Number of Revolutions for different Piston Speeds in Feet per Minute.
THE ECCENTRIC.
THE SLIDE-VALVE.
PROPORTIONS OF SLIDE-VALVES.
LAP ON THE SLIDE-VALVE.
Table showing Amount of "Lap" required for Slide-valves of Stationary Engines when the Steam is to be Worked Expansively.
LEAD OF THE SLIDE-VALVE.
FRICTION OF SLIDE-VALVES.
BALANCED SLIDE-VALVES.
COMPRESSION.
CLEARANCE.
AUTOMATIC CUT-OFFS.
SETTING VALVES.
HOW TO SET A SLIDE-VALVE.
SETTING OUT PISTON PACKING.
HOW TO REVERSE AN ENGINE.
DEAD CENTRE.
HOW TO PUT AN ENGINE IN LINE.
PROPORTIONS OF STEAM-ENGINES ACCORDING TO THE BEST MODERN PRACTICE.

Table showing Proper Thickness for Steam-cylinders of different Diameters.
THE INVENTION AND IMPROVEMENT OF THE STEAM-ENGINE.
SIGNIFICATION OF SIGNS USED IN CALCULATIONS.
DECIMALS.
Decimal Equivalents of Inches, Feet, and Yards.
Decimal Equivalents of Pounds and Ounces.
Useful Numbers in calculating Weights and Measures, etc.
Decimal Equivalents to the Fractional Parts of a Gallon or an Inch.
UNITS.
THE METRIC SYSTEM OF MEASURES AND WEIGHTS.
Metric Measures of Length.
Metric Measures of Surface.
Metric Measures of Capacity.
Metric Weights.
PUMPS.
STEAM-PUMPS.
BLAKE'S SPECIAL STEAM FIRE-PUMP.
WRIGHT'S BUCKET-PLUNGER STEAM FIRE-PUMP.
Dimensions of the Bucket-plunger Steam Fire-pumps.
PROPORTIONS OF STEAM FIRE-PUMPS.
PROPORTIONS OF BOILER FEED-PUMPS.
PROPORTIONS OF MARINE-PUMPS.
PROPORTIONS OF WRECKING-PUMPS.
PROPORTIONS OF MINING-PUMPS.
PROPORTIONS OF AIR-PUMPS.
PROPORTIONS OF TANK-PUMPS.
PROPORTIONS OF BREWERS' AND DISTILLERS' PUMPS.
THE KNOWLES' STEAM FIRE-PUMP.

CONTENTS.

Earle's Steam Fire-pump.
Directions for Setting up Steam-pumps.
The Atlas Steam Fire-pump.
Conde's Challenge Steam Fire-pump.
Holly's Rotary Steam Fire-pump.
Proper Method of Locating Steam Fire-pumps.
The Injector.
The Pulsometer.
The Hydraulic Ram.
Boilers of Steam Fire-engines.
Causes of Foaming in Steam-boilers.
Evaporation in Steam-boilers.
Internal and External Corrosion of Steam-boilers.
Rules.
Rule for Finding the Heating Surface of Steam-boilers.
Definitions as Applied to Boilers and Boiler Materials.
Table of Safe Internal Pressures for Iron Boilers.
Longitudinal and Curvilinear Strains.
Heat.
Latent Heat of Various Substances.
Caloric.
Combustion.
Entire Coal Productions of the World.
Spontaneous Combustion.
Table showing the Temperature at which different Combustible Substances will Ignite.
Steam.
Economy of Working Steam Expansively.
Central and Mechanical Forces and Definitions.
Mensuration of the Circle, Cylinder, Sphere, etc.

CONTENTS.

PROPERTIES OF THE CIRCLE.
Table containing the Diameters, Circumferences, and Areas of Circles from $\frac{1}{16}$ of an Inch to 20 Inches, advancing by $\frac{1}{16}$ of an Inch up to 10 Inches, and by $\frac{1}{8}$ of an Inch from 10 Inches to 20 Inches.
LOGARITHMS.
Table of Logarithms of Numbers from 0 to 1000.
HYPERBOLIC LOGARITHMS.
Table of Hyperbolic Logarithms.
RULES FOR FINDING THE ELASTICITY OF STEEL SPRINGS.
Table showing the Actual Extension of Wrought-iron at Various Temperatures.
Table deduced from Experiments on Iron Plates for Steam-boilers, by the Franklin Institute, Phila.
Table showing the result of Experiments made on different Brands of Boiler Iron at the Stevens Institute of Technology, Hoboken, New Jersey.
HOW TO MARK ENGINEERS' OR MACHINISTS' TOOLS.
TO POLISH BRASS.
SOLDER.
CEMENT FOR MAKING STEAM-JOINTS AND PATCHING STEAM-BOILERS.
JOINTS.
RELATIVE VALUE OF FOREIGN AND UNITED STATES MONEY.
Table showing the Load that can be Carried by Man and Animals.
Man or Animal Working a Machine.
Table of Coefficients of Frictions between Plane Surfaces.
Table of Friction Coefficients for different Pressures up to the Limits of Abrasion.

A CATECHISM

OF

High-Pressure or Non-Condensing STEAM-ENGINES;

INCLUDING

THE MODELLING, CONSTRUCTING, RUNNING, AND MANAGEMENT OF STEAM-ENGINES AND STEAM-BOILERS.

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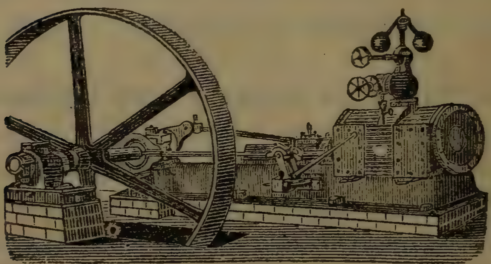


PHILADELPHIA:
E. CLAXTON & COMPANY.

32

ROPER'S CATECHISM

OF

STEAM ENGINES.

OPINIONS OF THE PRESS.

From the North American and United States Gazette.

A Catechism of High-Pressure Steam Engines, by Stephen Roper, published by E. Claxton & Company, Philadelphia. Mr. Roper, himself a practical engineer, has undertaken to furnish his fellow-engineers with the information experience has shown him to be most valuable. A number of tables of constant utility are furnished, and many rules and much practical advice. The work is plain rather than scientific in its language, and, claiming to be the only one expressly calculated for engineers, cannot fail to find quick demand and be of great value.

From the Scientific American.

A Catechism of High-Pressure or Non-Condensing Steam Engines, by Stephen Roper, Engineer. This is a valuable book on the steam engine. It contains much needed general information for engineers, as well as a description of many American improvements and specialties in steam engineering

CONTENTS.

INTRODUCTION.
THE STEAM-ENGINE.
WATER.
AIR.
HEAT.
THE THERMOMETER.
  Comparative Scale of English, French, and German Thermometers.
STEAM.
  Table showing the Temperature of Steam at different Pressures.
THE ENGINEER.
THE STEAM-BOILER.
  Cylinder Boilers.
  Flue Boilers.
  Tubular Boilers.
  Double-Deck Boilers.
  Locomotive Boilers.
  Mud-Drums.
  Boiler-Heads.
  Boiler-Shells.
  Steel Boilers.
  Internal and External Pressures.
  Rules.
  Table of Internal Pressures.
  Foaming in Steam-Boilers. Rust.
  Patent Steam-Boilers.
THE SAFETY-VALVE.
FEED-WATER HEATERS.
FUEL.
CHIMNEYS.

SMOKE.
GRATE-BARS.
DUTIES OF AN ENGINEER IN THE CARE AND MANAGEMENT OF THE STEAM-BOILER.
STEAM-ENGINES.
Table showing the Average Pressure of the Steam upon the Piston throughout the Stroke.
Lap on the Slide-Valve.
Table showing the Amount of "Lap" required for Slide-Valves when the Steam is to be worked expansively.
Lead on the Slide-Valve.
"Cushion."
Setting Valves.
Size of Steam-Port.
Size of Steam-Pipe.
Size of Piston-Rod.
Material for Different Parts of Engines.
Proportions of Engines.
Reversing an Engine.
Putting an Engine in Line.
Setting up Engines.
RULES FOR THE CARE AND MANAGEMENT OF THE STEAM-ENGINE.
DIFFERENT KINDS OF ENGINES.
KNOCKING IN ENGINES.
VACUUM.
THE INDICATOR.
THE GOVERNOR.
THE INJECTOR.
STEAM-PUMPS.
CENTRIFUGAL PUMPS.

NOISELESS BOILER FEED-PUMP.
Directions for Setting Up Steam-Pumps.
Table containing the Diameter, Circumferences, and Areas of Circles, and the Cubical Contents of Cylinders, in Gallons.
PISTON-ROD PACKING.
INCRUSTATION.
BOILER EXPLOSIONS.
STEAM- AND FIRE-REGULATOR.
CENTRAL AND MECHANICAL FORCES.
MENSURATION.
Circle, Cylinder, Sphere, etc.
BELTING.
Leather Belts.
Lacing Belts.
Horizontal Belts.
Perpendicular Belts.
Greasing Belts.
Rules for finding the Proper Width of Belts.
RULES TO BE OBSERVED IN CASE OF ACCIDENTS.
A BRIEF HISTORY OF THE STEAM-ENGINE.
History of the Different Parts of the Steam-Engine in Detail.
VOCABULARY OF TECHNICAL TERMS as applied to Different Parts of Steam-Engines and Steam-Boilers.
PROPORTIONS of Steam-Engines according to the best modern practice.
CEMENT for making Steam-Joints and patching Steam-Boilers.
HOW to mark Engineers' or Machinists' Tools.
To polish Brass.
NON-CONDUCTORS for Steam-Pipes and Cylinders.

THE

ENGINEER'S HANDY-BOOK.

CONTAINING

A FULL EXPLANATION OF THE STEAM-ENGINE INDICATOR, AND ITS USE AND ADVANTAGES TO ENGINEERS AND STEAM USERS. WITH FORMULÆ FOR ESTIMATING THE POWER OF ALL CLASSES OF STEAM-ENGINES; ALSO, FACTS, FIGURES, QUESTIONS, AND TABLES FOR ENGINEERS WHO WISH TO QUALIFY THEMSELVES FOR THE UNITED STATES NAVY, THE REVENUE SERVICE, THE MERCANTILE MARINE, OR TO TAKE CHARGE OF THE BETTER CLASS OF STATIONARY STEAM-ENGINES.

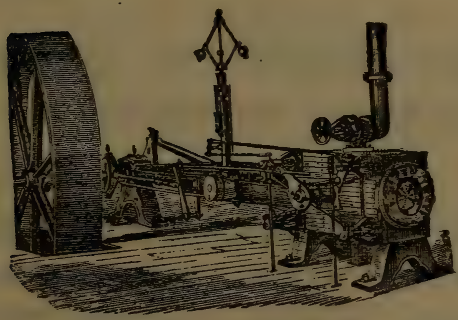
With Illustrations.

BY

STEPHEN ROPER, ENGINEER,

Author of

"Roper's Catechism of High Pressure or Non-Condensing Steam-Engines," "Roper's Hand-Book of the Locomotive," "Roper's Hand-Book of Land and Marine Engines," "Roper's Hand-Book of Modern Steam-Fire Engines," "Improvements in Steam-Engines," "Use and Abuse of the Steam-Boiler," "Questions for Engineers," etc., etc.



PHILADELPHIA:

E. CLAXTON & COMPANY.

4 37

This Book treats on every branch of

Steam Engineering

AND

Steam-Engines of Every Description

In use at the present day,—

CONDENSING, NON-CONDENSING,
SIMPLE, AND COMPOUND,

for Whatever Purpose Employed,
whether for

Engineering, Manufacturing, Pumping, Locomotion, Mining, Hoisting, or Propulsion,

AND IS MORE FULLY ILLUSTRATED THAN ANY OTHER WORK EVER HERETOFORE PUBLISHED ON THE SAME SUBJECT.

OPINIONS OF THE PRESS

ON

Roper's Engineer's Handy-Book.

The Manufacturer and Builder, New York.

An Engineer's Handy-Book.—Mr. Roper, the writer of this work, is well known to many of our readers as the author of a number of useful reference books relating to steam-engineering, which have become deservedly popular by reason of their plain, intelligible style, and their freedom from unnecessary and confusing mathematical technicalities. We would be glad to see Roper's hand-books largely multiplied and distributed in every workshop, for it is only out of books of this kind that the average workman will be able to master the principles of his handiwork.

Millstone, Indianapolis, Ind.

"The Engineer's Handy-Book," by Stephen Roper, Engineer, is a practical treatise on the management of the steam-engine. The author says the book was "not written for the purpose of instructing engineers how to design or proportion steam-engines or boilers, but rather to inform them how to take care of and manage them intelligently." The declaration is carried out in the plainest and most systematic manner.

As a text-book for students in mechanical engineering, it will be found of great value. Its illustrations and tabulated matter are important features, and printed in excellent style.

National Car-Builder, New York.

ROPER'S ENGINEER'S HANDY-BOOK.—This compact and comprehensive little volume contains a vast amount of information relative to the care and management of every class of steam-engines. It is profusely illustrated, and abounds in facts, figures, rules, tables, questions and answers, formulæ, etc., that are exceedingly valuable to engineers, and of easy reference by means of a copious and well-arranged index. The various subjects are discussed with brevity and clearness, and with a freedom from technicality which enables the reader to get at the pith of the matter without fishing it out from an ocean of words. A prominent feature of the book is a full explanation of the steam-engine indicator, and its use and advantages to engineers and others.

Leffel's Illustrated News, Springfield, Ohio.

ENGINEER'S HANDY-BOOK: By Stephen Roper, Engineer. —The author of the valuable series of hand-books which we have before referred to, has just issued the above-named work, which must find ready way into the hands of engineers and steam-users throughout the entire land. It contains a full explanation of the steam-engine indicator, its uses and advantages, with formulæ for estimating the power of all classes of steam-engines; also facts, figures, questions, and tables for engineers who wish to qualify themselves for the United States navy, the revenue service, the merchant marine, or the better class of stationary engines.

The American Engineer, Chicago, Ill.

THE ENGINEER'S HANDY-BOOK.—We are in receipt of the above work, which contains a description of the various forms of engines now in use, and supplies interesting and useful information as to the care, management, and remedy of defects of steam machinery and its appendages, with tables for calculating the power of engines.

American Machinist, New York.

ROPER'S ENGINEER'S HANDY-BOOK.—The subjects in this work have been treated in a brief and comprehensive way, therefore the reader is not required to read a number of chapters in order to acquire a little knowledge. The use of the indicator is treated in a plain, practical way, so that it may be readily understood. Abstruse formulas have been omitted and simple arithmetic used, thus avoiding the usual vexations among practical men who are uneducated in the higher mathematics. The author has in this book given the results of his own practical experience, which extends over a period of thirty years and upwards, and the work will doubtless be read with pleasure and profit by very many practical mechanics.

Engineering News, New York.

AN "ENGINEER'S HANDY-BOOK."—As a writer on subjects relating to steam and steam-engineering, Mr. Roper is now too well known to need any further introduction. In this, his latest contribution to steam-engineering literature, Mr. Roper has aimed to present to his brother engineers a "handy book" that will be to them what Trautwine's "Pocket-Book" is to civil engineers, and in doing this he has spared no labor in collecting and editing his materials. Some idea of the completeness of the work may be gathered from the statement of the publishers that it contains nearly 300 main subjects, 1316 paragraphs, 876 questions and answers, 52 suggestions and instructions, 105 rules, formulæ, and examples, 149 tables, 164 illustrations, 31 indicator diagrams, and 167 technical terms; over 3000 different subjects.

Boston Journal of Commerce.

MR. STEPHEN ROPER is well known as the author of several other handy-books that treat on steam, steam-boilers, and engines. This new work is, in our judgment, his best.

The Scientific American, New York.

A WELL made pocket-book of practical information for mechanical engineers, particularly those of limited education, and such as may wish to qualify themselves for service in the U. S. Navy or the mercantile marine. The more important engines in use are clearly described, and formulæ are given for estimating their power. Particular attention is paid to the steam-engine indicator, its use and advantages. The author has had much experience in this class of work, and writes clearly and plainly.

The Locomotive, Hartford, Conn.

ROPER'S ENGINEER'S HANDY-BOOK.—This last work of Mr. Roper is of special value to all who have to do with steam-boilers and engines, and it will be found a valuable shop companion for the mechanic. There are a great many facts collated that are not easily reached except through expensive books and libraries. These will be found of service to all classes of men, whether in trade or manufacturing. We commend it heartily, and believe it will have a large sale.

Forest, Forge, and Farm, Ilion, N. Y.

ENGINEER'S HANDY-BOOK.—We have received a book with the above title, by the well-known author and engineer, Stephen Roper, who has written a number of works on the subject of engineering. The eminent reputation of the author is a sufficient guarantee that the book is both interesting and useful. Mr. Roper has had an experience of over thirty-five years with all kinds of engines and boilers, and thoroughly understands locomotive, fire, marine, and stationary engines.

CONTENTS.

The Engineer.
Facts that should be Borne in Mind by Engineers.
Steam-Engineering as a Science.
Examination of Candidates for Cadet Engineers in the U. S. Navy.
Instructions how to Prepare for Examination for Engineer in the U. S. Navy and Revenue Service.
Instructions how to Obtain an Engineer's License in the Mercantile Marine Service.
Instructions how to Procure a License to Take Charge of Stationary Engines in any State or City requiring it.
The Steam-Engine INDICATOR; its Construction and Utility.
Different Kinds of Indicators.
Functions of the Indicator.
Technical Terms Employed in Connection with the Indicator.
How to Attach the Indicator.
Motion of the Paper on the Drum of the Indicator.

CONTENTS.

Most Correct Method of Adjusting the Indicator.
Most Reliable Parts of the Steam-Engine to which to Attach the Indicator.
How to Accurately Test the Attachments of the Indicator.
The Indicator Diagram.
The Explanatory Diagram.
The Theoretic Diagram.
The Actual Diagram.
Analysis of the Diagram.
Analysis of the Diagram Simplified.
Analysis of the Diagram made Easy.
Diagrams taken from Automatic Cut-off Engines.
Diagrams taken from Throttling Engines.
Diagrams taken from Compound Engines.
Diagrams taken from Simple Engines.
Diagrams taken from Locomotives.
Diagrams taken from Condensing Engines.
Instructions for Making an Analysis of Diagrams.
Instructions how to Space the Ordinates.
The Theoretical Expansion Curve.
Application of the Theoretical Expansion Curve.
How to Draw the Theoretical Expansion Curve.
How to Locate the Theoretical Terminal Pressure.
How to Calculate the Mean Effective Pressure.

How to Calculate the Theoretical Economy by the Diagram.
How to Calculate the Theoretical Rate of Water Consumption by the Diagram.
How to Make Allowance for Cushion and Clearance.
How to Estimate the Effective Compression.
What Indicator Diagrams Show.
The Planimeter.
Steam.
Superheated Steam.
Temperature of Steam.
Volume of Steam.
Surcharged Steam.
Evaporation of Steam.
Re-evaporation of Steam.
Latent Heat of Steam.
Sensible Heat of Steam.
Heat Necessary to Generate Steam.
The Quantity of Water Necessary to Condense a Certain Quantity of Steam.
Tables of Volumes of Steam for Different Pressures.
Weight of Steam.
Effluent Velocity of Steam at Different Pressures.
Steam Worked Expansively.
Steam-Jackets.
Steam-Domes.
Steam-Jets.
Steam-Chimneys.

CONTENTS.

Relative Volume of Steam at Different Pressures.
Relative Volume of Steam to the Water from which it was Generated.
Relative Quantity of Water Required to Condense Steam.
Steam Generated from Fresh and Salt Waters.
Condensation of Steam in Steam-Cylinders and Pipes.
Quantity of Steam Required for Heating Purposes.
Steam as a Means of Putting Out Fires.
The Expansive Properties of Steam.
Steam-Engines.
Peculiarities of Design and Construction of the Engines of the Different Lines of Steamships Plying between the Different Ports of this Country and those of Other Parts of the World, with Descriptions of the Same.
Peculiarities of Design and Construction of All the Different Automatic Cut-off Stationary Engines of this Country, with Descriptions.
The Difference between Automatic Cut-off and Throttling Engines.
The Advantages of Automatic Cut-off Engines over Throttling, and Vice Versa.
The Advantages of Large Steam-Engines over Small Ones, and Vice Versa.
The Advantages of Horizontal Engines over Vertical, and Vice Versa.

The Advantages and Disadvantages of the Different Steam-Engine Cut-offs, viz., the Automatic, Positive, Adjustable, and Riding.
A Description of All the Cut-offs in Use on Stationary and Marine Engines at the Present Day.
The Advantages of Different Cut-offs over Each Other.
The Causes Most Likely to Induce some Steam-Engines to Develop Less Power than they Ought to do, while Others would Develop More.
Proportions of All the Different Engines in Use at the Present Day, according to Accurate Scale.
The Two Classes of Engines in Most General Use in the World.
The Advantages of the Condensing over the Non-Condensing Engine, and Vice Versa.
The Difference between Simple and Compound Engines, their Advantages and Disadvantages.
How the Power of any Steam-Engine may be Increased within Certain Limits.
The Quantity of Fuel it will Require to Develop a Horse-Power in Different Engines.
The Quantity of Water that will Produce a Horse-Power in the most Improved Steam-Engines, as well as the Quantity Required for those of Inferior Type.
The Difference in Point of Economy between Condensing and Non-Condensing Engines.

CONTENTS.

The Difference in First Cost, Cost of Maintenance between Condensing and Non-Condensing Engines.
The Advantages and Disadvantages of Fast and Slow Speed Engines.
Why certain Types of Engines have been Abandoned, and Others Adopted.
High-Pressure Compound Engines.
Low-Pressure Compound Engines.
Schemes for Revolutionizing the Economy of Steam-Engines.
Steam-Engine Economy.
Instructions for Placing Steam-Engines in Steamships, Tug- and Ferry-Boats.
Instructions for Setting Up, Lining, and Reversing Stationary Steam-Engines.
Rules for Estimating the Power of Steam-Engines by Formulæ, and by Indicator Diagrams.
Rules for Finding the Right Size Engine to do a Certain Amount of Work.
Rule for Finding the Size of the Cylinder for an Engine of any Power, when the Pressure and Travel of the Piston are Known.
Rule for Finding the Quantity of Steam any Engine will Require.
Rules for the Care and Management of All Classes of Steam-Engines.
Steam-Engine Governors.
Slide-Valves.
Lap on the Slide-Valve.
Lead on the Slide-Valve.

How to Tell the Amount of Lap and Lead on a Slide-Valve without Opening the Steam-Chest.
Table Showing the Amount of Lap Required for Any Desired Cut-off.
Rule for Finding the Amount of Lap Necessary for Any Desired Cut-off.
How to Set a Slide-Valve Accurately.
Friction of Slide-Valves.
Balanced Slide-Valves.
Puppet-Valves.
Double-Beat Valves.
Throttle-Valves.
Relief-Valves.
Rotary-Valves.
Semi-Rotary or Oscillating-Valves.
Basket-Valves.
Gridiron-Valves.
Valve-Gear.
Releasing Valve-Gear.
Independent Valve-Gear.
Expansion Valve-Gear.
Reversing Valve-Gear.
Whole-Stroke Valve-Gear.
Vocabulary of Technical Terms as Applied to the Different Parts of the Valve-Gear of Steam-Engines.
Stop-Cocks, Valves, and Pipes for Whatever Purpose Employed in Connection with Steam-Engines.
Bed-Plates and Housings.
Steam-Cylinders.

Cylinder-Head Bolts.
Steam-Pistons.
Spring-Pistons.
Piston-Rods.
Stuffing-Boxes.
Steam- and Exhaust-Pipes.
Rock-Shafts.
Cross-Heads.
Eccentrics.
Cranks.
Crank-Pin Bearings.
Crank-Shaft Journals.
Keys, Jibs, and Straps.
Fly-Wheels.
The Link fully Illustrated and Explained.
Shifting Links.
Stationary Links.
Condensers, Surface, and Jet.
Proportions of Condensers.
Advantages and Disadvantages of Different Condensers.
Relative Quantities of Water Required for the Two Methods of Condensation.
The Injector Condenser.
Korting's Jet-Condenser.
The Vacuum.
How the Vacuum is Measured.
How the Vacuum is Maintained.
How the Vacuum is Produced.
The Effect of the Vacuum.
Air-Pumps.

Capacity of Air-Pumps According to best Modern Practice.
Relative Proportion of Air-Pump Cylinders to the Cylinders of Modern Steam-Engines.
Circulating Pumps.
Different Kinds of Circulating Pumps.
Relative Proportions of Circulating Pumps.
Marine Pumps.
Wrecking Pumps.
The Saliometer.
Barometer Gauges.
Thermometers.
Marine-Engine Registers.
Spring, Mercury, Syphon, and Vacuum Gauges.
Table of Rhombs, or Points of the Compass.
Technical Terms and Definitions Used in Navigation.
Tables of Knots and Miles as Measured by Various Nations.
Table of Leagues and Miles.
Length of the Day at Different Parts of the World.
Sailing Distance in Geographical Miles from New York to Different Points on the Globe.
Latitude and Longitude of Different Places.
Marine Signals.
Marine Bell, Whistle, and Light Signals.
Railroad Signals.
Pumps.
Feed-Pumps: their Capacity, etc.
Injectors: their Capacity, Efficiency, etc.

Screw-Propellers.
The Screw as a Means of Propulsion.
Different Kinds of Screw-Propellers.
Thrust-Blocks.
Stern-Tubes.
Paddle-Wheels.
Different Kinds of Paddle-Wheels.
Comparative Efficiency of Screw-Propellers and Paddle-Wheels.
Air: its Weight, Height, Effect, etc.
Table Showing the Weight of the Atmosphere at Different Altitudes above Sea Level.
Table Showing the Force of the Wind.
Table Showing the Relative Volume of Air at Different Temperatures.
Fuel.
Different Kinds of Fuel: their Comparative Value, etc.
The Chemical Constituents of Different Kinds of Fuel.
Heat.
Effects of Heat.
Capacity of Different Bodies for Heat.
Diameters, Circumferences, and Areas of Circles.
Metals and Alloys.
Marine Boilers.
Different Kinds of Marine Boilers.
Vocabulary of Technical Terms as Applied to Different Parts of Marine Boilers.
Vocabulary of Technical Terms as Applied to Different Parts of Marine Engines.

Super-Heaters.
Feed-Water Heaters.
Funnels.
Smoke-Stacks.
Air-Casings.
Blast-Pipes.
Spanner-Guards.
Meaning of the Term Mean Effective Pressure.
Meaning of the Term Average Pressure.
Difference between Mean Effective and Average Pressure.
Difference between Boiler Pressure and Pressure per Gauge.
Difference between Boiler and Cylinder Pressures.
Causes of Decrease of Pressure between Boilers and Cylinders.
Causes why Boiler Pressures do not Represent Cylinder Pressures.
The Probable Average Pressure in any Steam-Cylinder as Compared with the Boiler Pressure.
Why Boiler Pressures Cannot be Realized in the Cylinders of Steam-Engines.
Mistakes in Employing Boiler Pressures in Estimating the Power of Steam-Engines.
Tables of Circumferences, Diameters, and Areas of Circles from $\frac{1}{64}$ to 100 Inches.
Tables of Logarithms from 0 to 1000.
Tables of Hyperbolic Logarithms.
Use of Logarithms.
Utility of Logarithms

GEOMETRY.
TRIGONOMETRY.
MENSURATION.
NAVIGATION.
GEOGRAPHY.
NATURAL PHILOSOPHY.
AXIOMS.
THEOREMS.
PROPOSITIONS.
SOLUTIONS.
COROLLARIES.
TABLES OF SQUARES, CUBES, AND CUBE-ROOTS OF NUMBERS FROM 1 TO 1000.
MEANING OF THE TERM "CUBED."
MEANING OF THE TERM "SQUARED."
MEANING OF THE TERM "QUOTIENT."
MEANING OF THE TERM "PRODUCT."
ADDITION.
SUBTRACTION.
MULTIPLICATION.
DIVISION.
PROPORTION.
COMMON FRACTIONS.
DECIMALS.
TRIANGLES.
EQUILATERAL.
ISOSCELES.
SCALENE.
ACUTE.
OBTUSE.
RIGHT ANGLE, ETC.

THE CENTENNIAL CORLISS ENGINE.
WRIGHT'S AUTOMATIC CUT-OFF ENGINE.
THE WOODBURY, BOOTH & PRYOR'S AUTOMATIC CUT-OFF ENGINE.
DOUBLE-SLIDE VALVES.
SEMI-ROTARY VALVES.
THE BROWN AUTOMATIC CUT-OFF ENGINE.
THE HARRIS CORLISS ENGINE.
MARINE ENGINES.
MODERN MARINE COMPOUND ENGINES.
SECTIONS OF MARINE COMPOUND ENGINES.
SECTION OF SLIDE-VALVE ENGINE.
THE WOODRUFF & BEACH AUTOMATIC CUT-OFF HIGH-PRESSURE ENGINE.
EXPANSION GEARS.
THE PUTNAM MACHINE COMPANY'S AUTOMATIC CUT-OFF ENGINE.
THE GREEN AUTOMATIC CUT-OFF HIGH-PRESSURE ENGINE.
THE DOUGLASS AUTOMATIC CUT-OFF ENGINE.
THE BABBITT & HARRIS STEAM-PISTON.
PISTON, CONNECTING-ROD, AND CRANK-CONNECTION.
THE REYNOLDS CORLISS ENGINE.
THE CRANK.
THE LINK.
VALVE-GEARS.
THE WATERTOWN AUTOMATIC CUT-OFF ENGINE.
THE WATERS GOVERNOR.
THE SHIVE GOVERNOR.
REVERSING-GEAR FOR STEAM-ENGINES.
DIAGRAMS OF SLIDE-VALVE.

Wheelock's Automatic Cut-Off Engine.
Section of the Cylinder, Piston, Steam- and Exhaust-Valves of Wheelock's Automatic Cut-Off Engine.
Poppet Valves.
Slide-Valves.
The Wells Two-Piston Balance-Engine.
Section of the Wells Two-Piston Balance-Engine.
The Wardwell Valveless Engine.
The Steam-Engine Indicator.
Section of the Indicator.
Thompson's Indicator.
Richards' Parallel Motion Indicator.
The Atlas Corliss Engine.
Indicator Diagrams.
The Planimeter.
Diagram Measured by the Planimeter.
The Porter-Allen High-Speed Engine.
End View of a Surface Condenser.
The Injector Condenser.
Independent Condenser and Air-Pump.
Independent Air- and Circulating-Pump, with Air-Pump at one End, Circulating-Pump at the Other, and Steam-Cylinder in the Middle.
Section of Marine Air-Pump.
Independent Marine Circulating-Pump.
Marine Wrecking-Pump.
The Salinometer.
The Hotwell Thermometer.
The Uptake Thermometer.

Marine Steam-Engine Register.
Spring Steam-Gauges.
Marine Whistle Signals.
Marine Light Signals.
Marine Bell Signals.
Railroad Signals.
Pumps.
William Sellers & Co.'s Lifting Injectors.
Section of William Sellers & Co.'s Lifting Injector.
Rue's "Little Giant" Injector.
Friedman's Injector.
The Keystone Injector.
The Eclipse Injector.
The Clipper Adjustable Injector.
Section of Clipper Injector.
Mack's Fixed-Nozzle Injector.
The Inspirator.
The Ejector or Lifter.
Jamison's Steam Water-Ejector.
Water-Tubular Marine-Boiler.
Fire-Tubular Marine-Boiler.
Direct Flue and Return Tubular Marine-Boiler.
Method of Bracing Marine Steam-Boilers.
The Buckeye Automatic High-Pressure Cut-Off Engine.
Diagrams of Circles.
The Wetherill Corliss Engine.
Diagram of Steam-Joints.
The Fitchburg Steam-Engine.
The Fitchburg Governor.

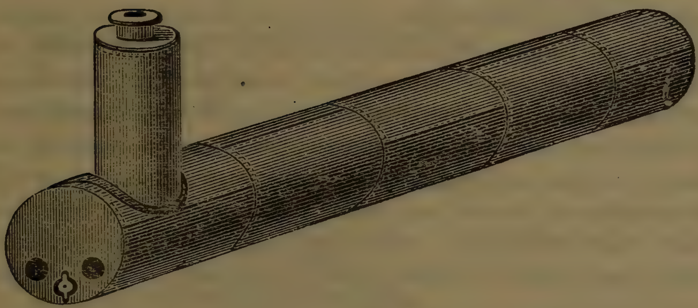
The Engineer's Handy-Book contains nearly 300 Main Subjects, 1316 Paragraphs, 876 Questions and Answers, 52 Suggestions and Instructions, 105 Rules, Formulæ, and Examples, 149 Tables, 195 Illustrations, 31 Indicator Diagrams, and 167 Technical Terms; over 3000 different Subjects, with the Questions most likely to be asked when under examination, before being commissioned as an Engineer in the U. S. Navy or Revenue Service; before being licensed as an Engineer in the Mercantile Marine Service, or receiving a certificate to take charge of a steam-engine or boiler in locations where such certificate is necessary. There is not a subject within the whole range of steam-engineering on which it does not treat.

With a great variety of other information not to be found in any other book ever published on the same subject in this country or in Europe, and more fully illustrated than any other work ever published on this subject.

USE AND ABUSE

C F

THE STEAM-BOILER.



BY

STEPHEN ROPER, ENGINEER,

Author of

"Roper's Catechism of High-Pressure or Non-Condensing Steam-Engines," "Roper's Hand-Book of the Locomotive," "Roper's Hand-Book of Land and Marine Engines," "Roper's Hand-Book of Modern Steam Fire-Engines," "Roper's Handy-Book for Engineers," "Roper's Improvements in Steam-Engines," "Questions for Engineers," etc.

PHILADELPHIA:
E. CLAXTON & COMPANY.

Use and Abuse of the Steam-Boiler.

OPINIONS OF THE PRESS.

Engineering News, Chicago, Ill.

Mr. Roper is the author of several well-known hand-books relating to the steam-engine, and steam machinery in general. In this, his latest work, he states that his object is, "simply to show what the results of his thirty years' personal experience with all classes of boilers prove to be the safest and most durable materials for their manufacture; to show the absolute necessity of good workmanship in their construction, and to call the attention of owners, engineers, and firemen to the rules that limit their usefulness, safety, and longevity." As in all his other hand-books, the writer addresses himself to men of ordinary intelligence,—those found in charge of steam-engines and boilers,—and in consequence his book is written in the plainest and most intelligible language that can be chosen. We have not the time, nor possibly the necessary amount of practical knowledge of all the latest improvements in steam-boilers, to criticise closely and intelligently the contents of the book, but in connection with it we would call attention to the large number of boiler explosions, attended with great loss of life, that have recently occurred in this country and in England, and which, upon investigation, have been proven to be the results of ignorance and carelessness on the part of attendants; and we cannot but think that steam-users would find it greatly to their advantage if such plain handy-books as those of Mr. Roper's were placed in the hands of every attendant upon a steam-boiler or engine, and his attention called to the advantage of making himself familiar with its contents.

CONTENTS.

Adjuncts of the Steam-boiler
Steam-boilers
Design of Steam-boilers
Forms of Steam-boilers
The Plain Cylinder Boiler
The Flue Boiler
The Tubular Boiler
The Double-deck Boiler
The Drop-flue Boiler
The Locomotive Boiler
Fire-box Boilers
Tubulous Boilers
Size of Boilers
Sectional Steam-boilers
Marine Boilers
Table showing the Number of Square Feet of Heating Surface to 1 Square Foot of Grate Surface in the Boilers of noted Ocean, River, and Ferry-boat Steamers
Boiler-heads
Steam-domes
Mud-drums
Water-space and Steam-room in Steam-boilers

Diameter and Length of Steam-boilers and Thickness of Boiler-plate
Evaporation in Steam-boilers
Evaporative Efficiency of Steam-boilers
Clapp and Jones' Vertical Circulating Tubular Boiler
Methods of Testing the Evaporative Efficiency of Steam-boilers
Proportion of Grate Surface to Heating Surface
Internal and External Corrosion of Steam-boilers
Internal Grooving in Steam-boilers
Silsby's Vertical Tubular Boiler
Expansion and Contraction of Boilers
Heating-surface of Steam-boilers
Rules for finding the Heating-surface of Steam-boilers
The Latta Steel Coil-boiler
Horse-power of Steam-boilers
The Moorhouse Safety Sectional Boiler
Setting Steam-boilers
Testing Steam-boilers .
Repairing Steam-boilers
Neglect of Steam-boilers
The Wiegand Sectional Boiler
Safe Working Pressure of Steam-boilers
Table of Safe Internal Pressures for Steel Boilers.
Table of Safe Internal Pressures for Iron Boilers.
The Roger's and Black Boiler
Selection of Steam-boilers.
Pulsation in Steam-boilers
Pierce's Rotary Tubular Boiler

CONTENTS.

LOCATION OF STEAM-BOILERS
THE HARRISON BOILER
BOILER-FLUES
Table of Squares of Thickness of Iron, and Constant Numbers to be used in finding the Safe External Pressure for Boiler-flues
Table of Safe Working External Pressures on Flues 10 Feet long
Table of Safe Working External Pressures on Flues 20 Feet long
COLLAPSING PRESSURE OF WROUGHT-IRON BOILER-FLUES $\frac{1}{4}$ INCH THICK
COLLAPSING PRESSURE OF WROUGHT-IRON BOILER-FLUES $\frac{5}{16}$ INCH THICK
COLLAPSING PRESSURE OF WROUGHT-IRON BOILER-FLUES $\frac{3}{8}$ INCH THICK
COLLAPSING PRESSURE OF WROUGHT-IRON BOILER-FLUES $\frac{7}{16}$ INCH THICK
THE SHAPLEY BOILER
BOILER TUBES
THE PHLEGER BOILER
Tables of Superficial Areas of External Surfaces of Tubes of Various Lengths, Diameters in Square Feet
Table of Superficial Areas of Tubes of different Lengths and Diameters from $2\frac{1}{2}$ to 3 Inches and from 8 to 20 Feet
STEAM-BOILER CONNECTIONS AND ATTACHMENTS.
GAUGE-COCKS
STEAM-GAUGES
GLASS WATER-GAUGES
THE BABCOCK AND WILCOX'S SECTIONAL STEAM-BOILER

CONTENTS.

Safety-valves
Table showing the Rise of Safety-valves, in parts of an Inch at different Pressures
Table of Comparison between Experimental Results and Theoretical Formulæ
Rules
Wittingham's Tubulous Boiler
Foaming in Steam-boilers
Incrustation in Steam-boilers
Prevention and Removal of Scale in Steam-boilers
Steam-boiler Explosions
Experimental Boiler Explosions
The Root Boiler
Vagaries of Experts in Regard to Steam-boiler Explosions
Defects in the Construction of Steam-boilers.
Improvements in Steam-boilers
The Allen Boiler.
Care and Management of Steam-boilers.
Instructions for Firing
Dampers
Steam-boiler Inspection
Rules for finding the Quantity of Water which Boilers and other Cylindrical Vessels are capable of Containing
Effects of different Kinds of Fuel on Steam-boilers
Boiler Materials
Steel
Strength of Iron Boiler-plate
Definitions as applied to Boilers and Boiler Materials

PUNCHED AND DRILLED HOLES FOR BOILER SEAMS.
Table showing the Strength of Welded Boiler-plates
PATENT BOILERS
THE GALLOWAY BOILER.
STRENGTH OF RIVETED SEAMS
COMPARATIVE STRENGTH OF SINGLE- AND DOUBLE-RIVETED SEAMS
HAND- AND MACHINE-RIVETING
COUNTER-SUNK RIVETS
RIVETS
Table showing Diameter and Pitch of Rivets for different Thicknesses of Plate
STRENGTH OF STAYED AND FLAT BOILER SURFACES
BOILER-STAYS
STAY-BOLTS
CALKING
TESTING-MACHINES
FEED-WATER HEATERS
Table showing the Units of Heat required to Convert One Pound of Water, at the Temperature of 32° Fah., into Steam at different Pressures
GRATE-BARS
CHIMNEYS.
Table showing the Proper Diameter and Height of Chimney for any kind of Fuel
Table showing Heights of Chimneys for producing certain Rates of Combustion per Square Foot of Area of Section of the Chimney
SMOKE
CONTRIVANCES FOR INCREASING DRAUGHT AND ECONOMIZING FUEL IN BOILER FURNACES

Table showing the Actual Extension of Wrought-iron at various Temperatures . . . .
Table showing the Linear Dilatations of Solids by Heat . . . .
Table deduced from Experiments on Iron Plates for Steam-boilers, by the Franklin Institute, Philadelphia . . . .
Table showing the Results of Experiments made on different Brands of Boiler Iron at the Stevens Institute of Technology, Hoboken, N. J. . .
Table showing the Weight of Cast-iron Balls from 3 to 13 Inches in Diameter. . . . .
Table showing the Weight of Cast-iron Plates per Superficial Foot as per Thickness . . .
Table showing the Weight of Round-iron from ½ an Inch to 6 Inches Diameter, One Foot Long.
Table showing the Weight of Boiler-plates One Foot Square and from $\frac{1}{16}$th to an Inch Thick .
Table showing the Weight of Square Bar-iron from ½ an Inch to 6 Inches Square, One Foot Long .
Table showing the Weight of Cast-iron Pipes, One Foot in Length, from ¼ Inch to 1¼ Inches Thick, and from 3 to 24 Inches Diameter. .
Table showing the Tensile Strength of various Qualities of American and English Cast-iron .
Table showing the Tensile Strength of various Qualities of American Wrought-iron. . .
Table showing the Tensile Strength of various Qualities of English Wrought-iron
To Polish Brass . . . .
Cement for making Steam-joints
Steam-dampers
Index

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Woodbury, Booth & Pryor, ... 2

Edson Recording Gauge and Alarm, ... 3

Southwark Foundry, ... 3

Knowles' Patent Steam Pumps, ... 4

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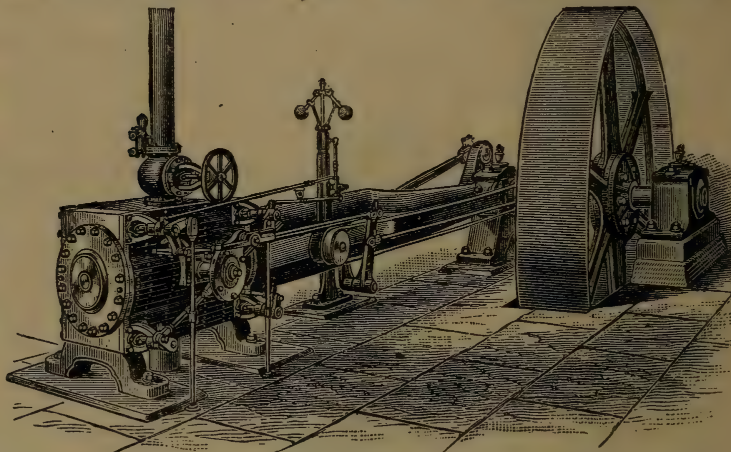

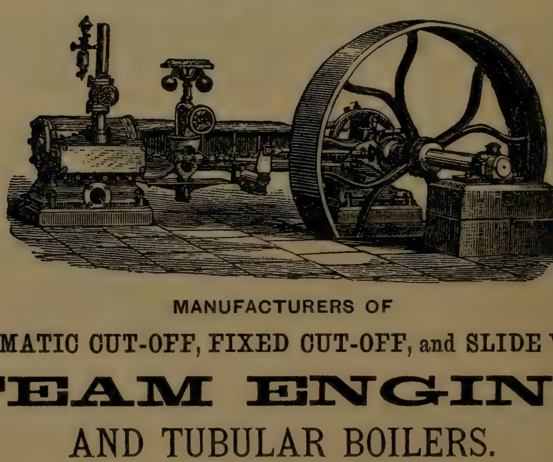

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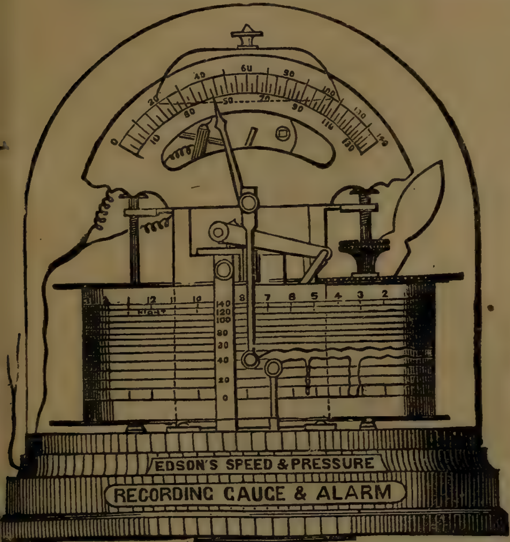

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